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SOLID LUBRICATED ROLLING ELEMENT BEARINGS SEMIANNUAL STATUS REPORT NO. 1

(Reporting Period: 1 August 1978 - 31 January 1979)

15 FEBRUARY 1979

MICHAEL N. GARDOS

Sponsored by:

Defense Advanced Research Projects Agency (DoD)
DARPA Order NO. 3576
Monitored by AFML/MBT Under Contract No. F33615-78-C-5196

Submitted by:

Hughes Aircraft Company
Technology Support Division
Culver City, CA 90230

AEROSPACE GROUPS
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Approved:
R. [illegible]. Sca[illegible]e, Manager
Advanced Technology
Laboratory

FOREWORD

This report was prepared by Hughes Aircraft Company under DARPA Order No. 3576, AFML Contract No. F33615-78-C-5196, effective starting date 1 August 1978, under the title of "Solid Lubricated Rolling Element Bearings." The work was administered under the technical direction of the Lubricants and Tribology Branch, Non-Metallic Materials Division of the Air Force Materials Laboratory, with Mr. B. D. McConnell acting as Project Engineer.

The Program Manager and Principal Investigator is Mr. Michael N. Gardos, (213) 391-0711, extension 4532.

This reporting period should be considered as the major part of the program definition stage. Technical work considered essential and unaffected by any reasonable program plan change in the near future was also performed.

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I. INTRODUCTION

This program is undertaking the complex effort of first proving the feasibility of advanced solid lubricated rolling element bearings for urgently needed extreme environment military applications, and then developing the necessary technology for the design, construction and testing of such bearings for use in the actual weapon systems. The key weapon system considered is the cruise missile and the specific bearing types slated for research and development are the cruise missile gyro (Type I) and turbine engine (Type II) bearings.

As determined by a unique philosophy, the program design has three major advantages:

1. From the onset, (a) the tribological fundamentals of bearing material and solid lubricant system interactions are investigated, along with (b) computer prediction of solid lubricated bearing dynamics, and (c) thoroughly instrumented bearing tests to check those predictions, and allow their appropriate modifications. The simultaneous performance of the above three research nodes completes the Triangle of Balanced Research (TBR), depicted in Figure 1. We consider the TBR as the only viable tool capable of assisting us to achieve the major initial program goal: correlating theory and practice of high risk research.

2. The stepwise progression of all phases assures maximum technology transfer from one phase to another: the science and technology of solid lubricated gyro bearings will serve as the preamble to that of the more demanding turbine engine bearing. At the same time the solid lubed gyro concept will indicate feasibility of advanced solid lubricated rolling element bearings and comprise a significant development in its own right.

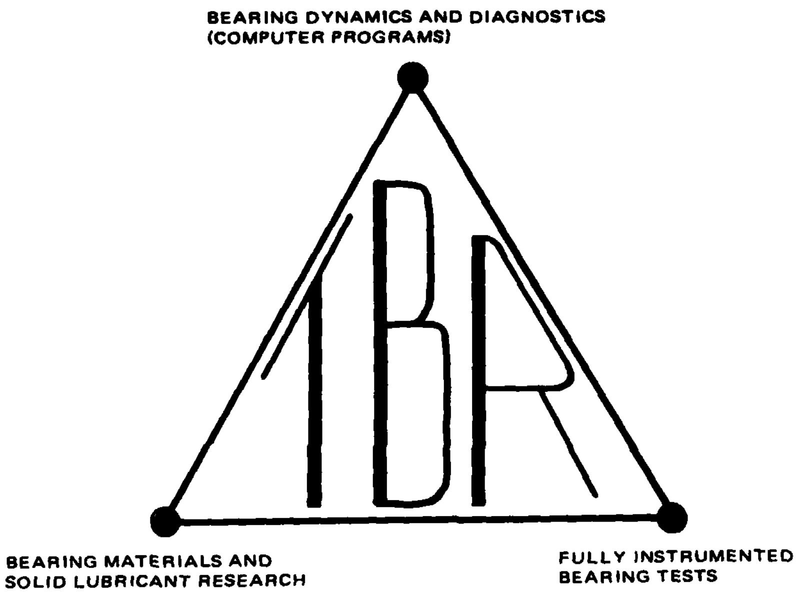

Figure 1. The Triangle of Balanced Research (TBR) concept, as applied to the DARPA/AFML/Hughes Solid Lubricated Rolling Element Bearing Program.

3. All research and development work is conducted by Hughes and selected subcontractors (U.S. as well as western European firms), forming an interdisciplinary work team of experts, specifically selected by Hughes to accomplish all of the above goals.

This program is run in two phases. The first phase (comprised of the end of FY'78, and the years of FY'79 and FY'80) is the feasibility study portion. It will follow a theoretical and experimental approach, where competitive, state-of-the-art bearing ball and race machining methods, intermetallic hardcoat/low shear strength solid lubricant softcoat systems, specially designed polymeric composite ball bearing retainers and bearing post treatments are considered for gyro bearings at the present time and for turbine engine bearings in the near future. Bearing operational parameters and materials performance parameters will be generated by bearing tests, by the use of a specially constructed ball bearing simulator and other,

simpler friction and wear testers. These parameters will be incorporated into computer programs to predict the dynamic behavior of a bearing as a "mini" system in itself and approximate its behavior in a larger system of a moving mechanical assembly (e.g., the bearing testers, the gyro assembly or the entire cruise missile turbine engine). The appropriate iterations will provide to the tribologist guidance as to the type of new solid lubricant developments needed in terms of thickness, traction, wear life, hardness, etc., and to the bearing designer tips in altering bearing configuration. The latter is especially important, because during the feasibility study portion we will attempt solid lubrication of bearings that were originally designed for operation with liquid or grease lubricants. It is not reasonable to hope that bearing design and running parameters (e.g., contact angle, race compliance, relative hardness of bearing components, bearing size, cage design, self-cooling features) remain identical for both liquid and solid lubricated rolling element bearings.

The second phase (FY'81 and FY'82 and any extension thereof) will consist of the full development of the two bearing types, keeping the TBR concept in mind at all times. Continuance into the second phase will be predicated on a Critical Design Review (CDR), held at the end of FY'80.

Selection of the cruise missile as the weapon system of central interest provides us with the ideal test vehicle, whose relatively short operational lifetime extends the promise of early feasibility for the scientifically upgraded versions of today's typically short lifetime, solid lubricated rolling element bearings. Limiting the top temperature limit for the turbine engine bearing to 500°F during the first phase of the program further aids our chances in achieving the same.

II. PROGRAM HISTORY

Many current and future weapon systems are being designed to operate for longer periods of time in increasingly severe environments. These ambitious mission objectives impose stringent requirements on the materials and design engineer. Of special concern is the design of and materials selection for moving mechanical assemblies, such as rolling element bearings, that drive components and equipment critical to the successful functioning of so many satellites, strategic ballistic missiles, aircraft, tactical and strategic cruise missiles and other auxiliary equipment.

Most, if not all, of the current designs relative to these moving systems are based on the use of conventional oil and grease lubricated assemblies which give designers serious reservations over their ability to meet the new system design life goals.

Oils and greases tend to (a) thicken or freeze at low temperature; (b) decompose, vaporize and oxidize at high temperatures; (c) migrate, separate (oil from additives), collect dirt and debris over long storage times; (d) present design limitations in terms of size and weight because of need for bulky oil reservoirs in case of engine applications; (e) exhibit complex degradation mechanisms in bearings which inhibit the development of reliable life prediction methods.

In general, solid lubricants overcome most all limitations of oils and greases. The inherent greater chemical stability of solid lubricants offers the potential for low (cryogenic) and high temperature performance (eventually up to 1500°F), long storage stability (10+ years), size and weight reduction (since oil reservoirs are unnecessary) and eventually improved life prediction (since the degradation mechanisms are fewer and simpler than with oils and greases).

The above problem and the likely solutions are, however, difficult and multifaceted. They led directly to the urgent need for this program, originally proposed by Hughes at a January 1978 presentation to DARPA.

The sequence of events from that first contact is described below:

January '78:	Presentation to DARPA/AFML/AFSC.
February '78:	Formal proposal submitted to DARPA.
March '78:	DARPA/AFML/AFSC accept proposal.
April to July '78:	Preparation of technical details and preliminary statements of work with first group of potential subcontractors.
July '78:	Program review by DARPA Materials Research Council (MRC); criticism by MRC on apparent lack of understanding of fundamentals.
August '78:	Hughes demonstrates realistic goals and soundness of basic approach to DARPA; program start date 1 Aug. '78.
September '78:	Hughes policy letter to DARPA/AFML/AFSC on details of fundamental approach; preparation for Preliminary Design Review (PDR); key subcontractors under contract negotiations.
October '78:	Intended subcontractors, potential subcontractors, government personnel and guests attended a successful two-day PDR to present and critique existing program plans; Hughes Program Office evaluates presentations and post PDR DARPA comments and redesigns program plan; Litton (Type I bearing subcontractor) representative is in western Europe preliminarily assessing solid lubricant and bearing technology there; Hughes Program Manager visits additional subcontractors and works out preliminary agreements on cooperation.

November '78: Hughes Program Manager presents revised program plan to DARPA/AFML/AFSC executive panel. DARPA accepts final program plan and the suggestion in include certain aspects of western European technology to break bottleneck of shortcomings on certain key technical areas not solvable with current U.S. state-of-the-art. Letter proposals received from all but two potential subcontractors. Battelle and Hughes agree on the final design of the ball bearing simulator; Litton, Barden, Battelle and MTI are under letter contracts; Hughes Program Manager and Litton representative begin extensive preparation for second fact-finding trip to Europe. Hughes subtask program plans for gyro bearing retainer preparation and softcoat deposition work begin.

December '78: Battelle builds successful brass-board model of ball bearing simulator. Hughes gyro retainer preparation and softcoat deposition plans finalized and work begins. Preparations for European trip itinerary completed. Hughes letter directive for the preparation of cylindrical 3D carbon/graphite weave preforms for turbine engine (-like) bearing retainers is sent to Fiber Materials, Incorporated, a potential subcontractor.

January '79: European survey trip successfully completed. Three cooperating Swiss firms can deliver hardcoated gyro bearing races and balls for Litton feasibility study. One French firm developed a three phase Fe/Mo/S self-lubricating compact that appears more desirable than MoS_2. Hughes purchase order to LSRH (Switzerland) is completed for bare steel and chemical vapor deposition (CVD) hardcoat bare steel and cemented carbide R-3 gyro bearing balls. Litton's first major report on the theory and practice of forthcoming solid lubricated gyro bearing research (including thorough technical trip report

January '79: (Continued)	dealing with recent European trip) is under preparation and due to Hughes Program Manager during the month of February 1979.

III. ACCOMPLISHMENTS TO DATE

A. Current Program Plan

Inasmuch as the program is still undergoing some modifications, the current program network is nearly, but not all, complete, as shown in Figure 2. Finalization depends on further negotiations with DARPA and the involvement of other, additional subcontractors.

As indicated in Figure 2, the majority of the technical thrust is directed toward the development of the solid lubricated turbine engine (Type II) bearing or, to be more precise, its lower temperature (500°F) precursor (major subcontractor, Mechanical Technology, Incorporated, Latham, New York; responsible engineer, Dr. Pradeep Gupta). MTI will be aided by SKF, Incorporated, King of Prussia, Pennsylvania (responsible engineer, Dr. Juris Pirvics), mainly in terms of computer prediction of bearing systems dynamics. Due to the reduced temperature requirements, some of the solid lubricant coating systems and the polymeric composite retainers developed for the Type I bearing should be applicable to the Type II bearing also. As previously explained, the solid lubricated version of a gyro (R-3) bearing will be shown feasible first (major subcontractor, Litton Guidance and Control Systems, Woodland Hills, California; responsible engineer, Mr. Willi Baginski), and will serve as a technological lead-in to the Type II bearing. It is, of course, a major development on its own merits.

Preliminary research indicated that: (a) the Type I (and Type II) bearing machining methods currently utilized are inadequate for use, and (b) the necessary machining techniques needed for smear and damage free ball and race surfaces do not presently exist. Moreover, the cost and lead time for a complete machining study is far greater than allowed by any reasonable version of our program plan. Therefore, a hard, intermetallic coating process had to be uncovered that exhibits minimum sensitivity to machining smear (i.e., provide maximum coating adhesion to the improperly

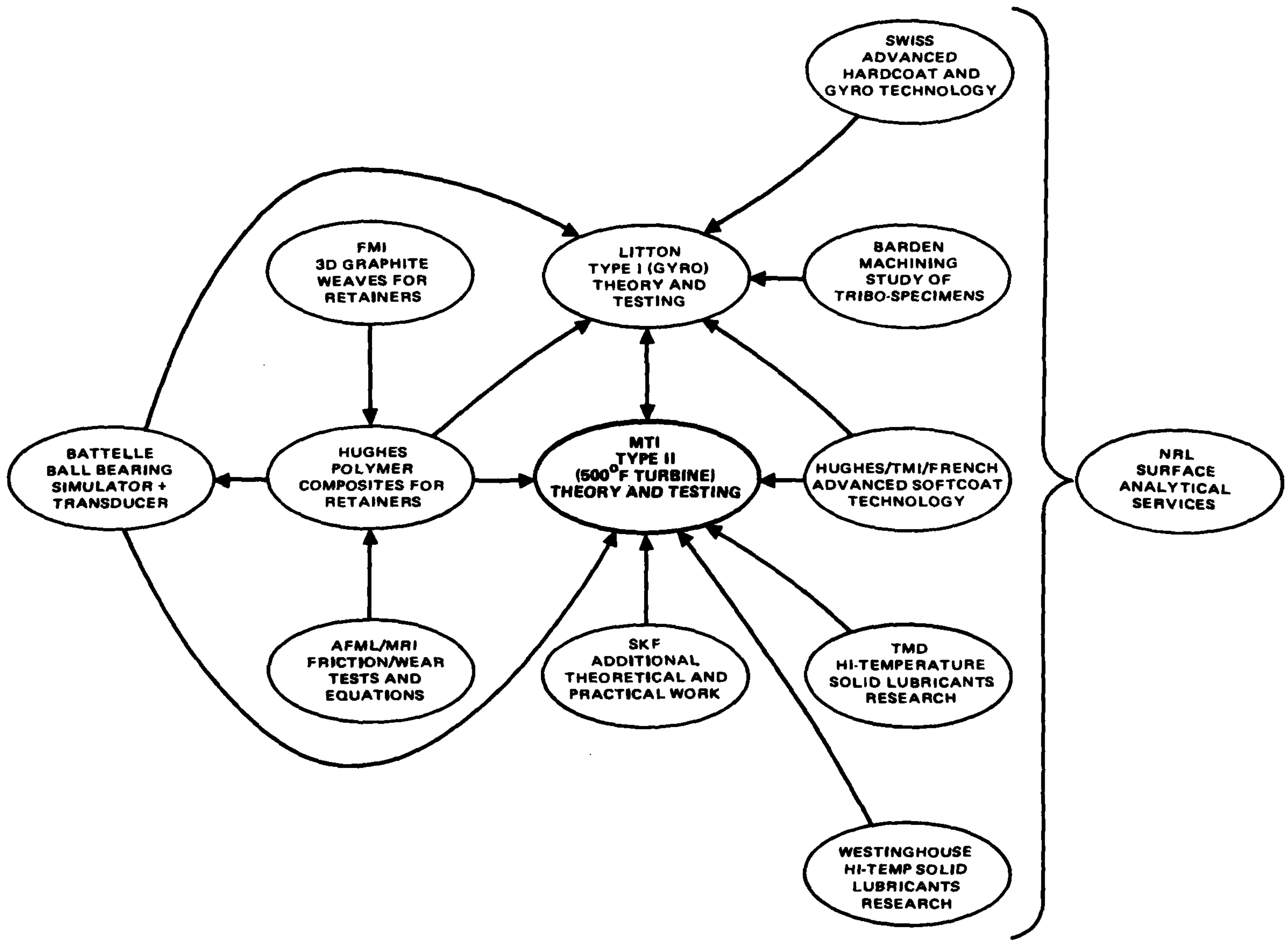

Figure 2. The current program network.

machined bearing steel). Selected cooperating Swiss firms were found to be the only ones able to fulfill these requirements at this time, through CVD hardcoating of Type I steel balls, steel races and cemented carbide balls with selected hard intermetallic compounds [purchased from the coating firm LSRH, Neuchatel, Switzerland; other participating firms are RMB, Biel-Bienne, Switzerland (final lapping of hardcoated races) and Saphirwerk, Nidau, Switzerland (lapping of hardcoated balls)]. Similar U.S. technology for the Type I bearing, or any such technology anywhere for the Type II bearing, are not yet available. The low shear strength, softcoat applications will be performed by Hughes Aircarft Company, Culver City, California (responsible engineer, Mr. Ronald I. Christy) and TMI, Santa Barbara, California (responsible engineer, Mr. Richard P. Riegert). Possible soft-coating support may be received from HEF, St. Etienne, France (responsible engineer, Mr. Antoine Gaucher).

The Hughes-developed polymeric (polyimide) composite retainers (top temperature limit, 600°F) for both bearing types are under development by Hughes Aircraft Company, Culver City, California (responsible engineer, Mr. Michael N. Gardos), with reinforcing composite preforms from a 3D carbon/graphite weaving firm (FMI, Biddeford, Maine; responsible engineer, Mr. Alden Lewis) and special wear test/wear equation work by AFML/MRI cooperative research, Wright-Patterson AFB, Ohio (responsible engineer, Mr. Karl Mecklenburg of MRI).

A special ball bearing simulator is being developed by Battelle Columbus Laboratories, Columbus, Ohio (responsible engineer, Mr. Jerrold Kannel) as an intermediate tool for testing (screening) materials under bearing-like operational conditions and generating parameters that lend themselves to substitution into dimensionless computer programs. The bearing simulator data should be equally applicable to the development of both bearing types.

Despite the fact that the turbine bearing temperature during the feasibility study portion of the program is reduced to 500°F, we are not losing sight of the final temperature limits. After the feasibility study, this limit will be increased first to 1000°F, then to 1500°F. The latter should be the ultimate top temperature limit for any turbine engine bearing planned for the foreseeable future. Consequently, fundamental research will be started on the theory and practice of intercalation compounds, using certain metal dichalcogenides as model compounds (TMD, Evergreen, Colorado; planned responsible engineer, Professor Warren Jamison) and the identification of the Ga/In/WSe_2 Westinghouse composite, in terms of crystal structure, stoichiometry and tribological behavior (Westinghouse R&D Center, Pittsburgh, Pennsylvania; planned responsible engineer, Mr. David Boes). These solid lubricants will withstand at least 1000°F, and the understanding of their physical-chemical fundamentals can lead to versions which can operate at even higher temperatures.

One of the most pressing technical problems remaining unanswered is the machining of bearings (both types) in a surface smear free condition. While the Swiss hardcoat process is somewhat forgiving to the smeared condition of small Type I bearing components, it cannot be utilized at this time for the larger Type II bearing parts because of the excessive distortion of the anisotropic bearing steels under coating temperature and re-heat treat temperature conditions (see forthcoming technical discussion). Since the needed bearing machining studies are deemed to be beyond the scope of the present program (aside from some initial efforts to provide the best bearing specimens possible using existing fabrication techniques or minor modifications thereof), separate parallel studies should be started. The development of more isotropic, homogeneous bearing steels is also needed. In order, however, to at least delve into the fundamentals of refractory hard metal coating behavior on smear-free versus smeared bearing steel surfaces of simple geometry, Barden Corporation, Danbury, Connecticut (responsible engineer, Mr. John J. Murphy), is conducting a limited machining study using flat friction and wear test specimens.

Since nearly all bearing machining and solid lubrication techniques are surface physics and chemistry dependent, all crucial research steps will be directed and controlled by modern surface analytical techniques, such as ESCA-SAM and ISS-SIMS methods. DARPA has made the analytical services of NRL, Washington D.C. (responsible engineer, Dr. James K. Hirvonen) available to our program, free of charge. The program, in turn, will evaluate the tribological behavior of special friction and wear test specimens ion-implanted by NRL.

The above overview of the program network will now be supplemented by the delineation of the technical achievements for the first six months of the program.

B. Technical Achievements

Since no major subcontractor reports have been received yet at the time of this writing, the following paragraphs represent an executive summary of the technical work to date. Also, to outline plans for forthcoming research, previous proposal inputs of present and potential subcontractors are utilized. Occasionally, references are made to subcontractor report(s) under preparation which will contain data and interpretation in greater detail.

1. Gyro (Type I) Bearing Research (Reference 2)

(a) Bearing Surface Damage by Machining Operations - Litton researchers found that fluid lubricated gyro bearing performance could not be improved by upgraded design techniques (e.g., computer simulations) beyond the 80 percent perfection limit. The last twenty percent appeared to be material (i.e., bearing component material) dependent and, specifically, to be a direct function of the physical metallurgy and chemistry of the raceway and the ball surfaces. A Litton developed layer etch technique, capable of removing about 15 - 30 Å layers of the steel bearing surface at a time, combined with scanning electron photomicrography, revealed that both the ball path and ball surfaces contained damage layers, formed by the respective machining processes. These layers are multifaceted: a thin, soft cover

layer (the Beilby smear) can hide as many as two subsurface damage regions. The one immediately below the Beilby layer consists of preferentially formed carbide particles far richer in concentration than the carbide distribution within the steel matrix. The cause of formation is hypothesized to be the pressure of grinding or lapping and the concomittant temperature increase. The decarburization of the nearby martensitic structure causes significant softening and provides the stock for the formation of the Beilby top smear. (Note that this preferential carbide formation occurs with low as well as high chromium steels, such as 52100 or 440C alike, and can happen during running of the bearing also.) The martensitic structure below the carbide particle layer is also distorted to various degrees, and occasionally can be destroyed almost beyond recognition.

Due to differences between the machining techniques of balls and races, the respective thickness, chemistry and morphology of the damage layers are not the same. Preferential subsurface carbide precipitation is far worse in the case of balls. The apparently excessive pressure of the lapping plates and the temperature increase of the ball surfaces during the prolonged (several hours) lapping process accelerates carbide formation. Since the duration and tool pressure of race grinding and lapping are somewhat different (i.e., less), carbide formation also occurs here, but not to the same extent as in the previous case. In both cases, however, the soft Beilby layer is easily machinable, providing the type of visual and instrument-measurable surfaces finishes for which quality control inspectors look. Unfortunately, the Beilby layer serves as a kind of undesirable and effective camouflage, hiding the damaged layers below.

During liquid lubricated gyro bearing operation (unit load = 150-200 Kpsi, 25,000 rpm rotation), the elasto-hydrodynamic (EHD) film is thin enough to cause some lubricant starvation. The poorly adhering Beilby layer (the interface between that and the carbide precipitation layer is filled with subsurface cracks and caverns partly permeated with the original machining and cleaning fluids) quickly fatigues and torn off, generating extremely high flash temperatures. These temperatures degrade

the hydrocarbon lubricant into uneven patches of varnish. This varnish, along with the file-like action of the exposed ball carbides on the relatively softer races, can cause gyro failures in running times as short as 50 hours. On the other hand, depending on some fortuitous combination of unintentionally ideal machining parameters for isolated batches of races and balls and through the accidental matching of good balls and races, gyro bearings have been known to run failure-free up to 40,000 hours. Yet the bearing component quality control engineer, who checks for surface flaws by optical microscopy, along with surface finish and out-of-roundness by stylus techniques, pronounces both the good and subsurface damaged bearings excellent prior to assembly and test.

Checks of specimens for numerous U.S. gyro bearing manufacturers and large roller and turbine bearing fabricators revealed this universal problem. Our recent European trip was prompted partly by the hope of finding ball bearing firms there who have identified this problem and have conducted research to combat it. As it turned out, with the exception of one Swiss ball manufacturer (see Reference 2) who had exhibited some understanding of the matter and intuitively adjusted the ball lapping methods to minimize "ball misbehavior", all other European firms were just as unaware of this problem as their U.S. counterparts. Nevertheless, Litton will examine numerous additional ball bearing specimens from both western European and other U.S. manufacturers, hoping to identify fabrication conditions that could yield smear-free surfaces.

It is very clear that presently available bearing surface conditions may be acceptable (or marginal) for fluid lubricated gyro operation, but would be dangerously unstable for substrates under solid lubricant coatings.

(b) Bearing Surface Stabilization by Refractory Hard Metal Coatings - As previously described in Reference 1, our preconceived idea of successful solid lubrication and corrosion protection of steel gyro (and turbine) bearings involves utilization of a layered coating system. First,

an adherent, compatable, refractory compound layer is applied directly to the metal alloy (or ceramic) bearing surfaces to stabilize them both chemically (i.e., to prevent environmental corrosion during storage or operation and preferential subsurface carbide formation during running of the bearing) and physically by possible alteration of some of the metallurgical damage induced by the machining process. This improvement occurs through desirable interface reactions. In addition, a desirably hard substrate (i.e., harder than the bearing steel) is provided for a thin, low shear strength top film of a layered-lattice solid lubricant. This hard substrate must adhere under Hertzian stresses of up to 150-200 Kpsi in the case of Type I and over 250 Kpsi with Type II bearings.

To attain the above goals, Litton extensively studied the variations in glow discharge sputtering techniques for thin film coating applications, e.g., diode and triode sputtering with or without substrate bias, ion plating, plasma anodization and reactive sputtering.

In summary, it was concluded that the sputtering processes were developed predominantly for electronic applications. The major aim of the electronic industry is to apply true stoichiometric compounds by sputtering techniques, where a strong interface development by diffusion is unwanted because of deviations and structural and, therefore, electronic changes in thin film composition. For the improvement of friction and wear of high speed, low and high load rolling element bearings, thin film stoichiometric compounds with limited adhesion do not apply. Moreover, improperly administered sputter cleaning techniques (e.g., backsputtering bearing hardware with Ar^+ ions) themselves can generate temperatures high enough to cause preferential carbide formation. The concomittant decarburization (i.e., adhesion-reducing softening) of the martensitic matrix base can further aggravate the high load delamination problem of a subsequently sputtered, thin, intermetallic hardcoat.

In order to obtain the strong interface and complex compound development by interdiffusion that leads to hardcoats with

improved adhesion, ductility, friction and wear, processes such as Chemical Vapor Deposition (CVD) and plasma activated versions of the same are required.

The thin film approach of the available sputtering processes can, however, be used for the application of the layer-lattice and other solid lubricants on bare or hardcoated ball bearing surfaces.

Independent literature and industry surveys by Litton and Hughes revealed (see Reference 1) that certain Western European research organizations, in cooperation with local ball bearing manufacturers, pioneered the work involving CVD chromium carbide (CrC)-titanium carbide (TiC) composite hardcoat/MoS_2 softcoat combinations in high speed instrument bearings (Reference 3). This and other encouraging information prompted a European fact-finding visit by the Litton responsible engineer, at which time CrC-TiC coated 440C instrument bearing races were brought back for examination. Note that this hardcoat system can be applied to both 440C and 52100 bearing steels.

Optical photomicrography and SEM/EDX examination of regular and tapered cross sections of these specimens by Litton and Hughes revealed the chemistry and physics of high adhesion. The results prompted a second visit by the Litton representative and the Hughes Program Manager to the key Swiss firms (LSRH, RMB and Saphirwerk) and other locations to lay the groundwork for obtaining hardcoated gyro bearing balls and races for further Litton research.

The initial samples consisted of CrC-TiC coated inner and outer races of R-3 bearings, both in the annealed-coated and the coated - quenched - tempered - lapped condition to define structural integrity and composition of the coating. Optical photomicrography of the ball path surfaces revealed no cracks or other defects on any of the coated parts.

Proper bonding to the steel interface is achieved by first depositing a thin CrC layer, topped by a layer of TiC (thickness ratio ≅ 1:3), onto fully hardened and precision machined races and balls, using a conventional CVD process with the general exchange reaction of

$$MCl_4(g) + CH_4(g) \rightarrow MC(s) + 4HCl(g).$$

Good adhesion is provided by the acid vapors that etch (i.e., clean) the bearing surfaces prior to full development of the first CrC layer and the high deposition temperature of the process itself (see Figure 3), which provides chromium and iron diffusion from the steel matrix into the CrC. This diffusion, which also permeates the TiC layer, was shown by EDX analysis at eight locations across the entire composite coating thickness (approximately 5 μm thick). The first location was in the steel substrate near the interface of the coating, while the others were placed at regular intervals throughout the CrC-TiC layer. Chromium and iron diffused through the TiC layer, exhibiting a concentration decrease gradient to the edge (top) of the coating. The same diffusion process was observed with titanium, which cross-diffused through the CrC interlayer into the steel substrate. The additional heating cycle before the requenching operation of the coated-annealed parts (see Figure 3) further enhances this diffusion, which eventually results in the formation of a graded, complex solid solution throughout the entire layer. This coating system provides unsurpassed adhesion and ductility at the steel/coating interface. It is known that brittle hard metal carbides can be made more ductile by alloying or interdiffusion by hard metal species.

An attempt was made to identify the stoichiometry of the graded carbides by ESCA-SAM at NRL prior to the second European trip. However, due to the untimely inoperability of the NRL Auger spectrometer, this work had to be postponed to a later date.

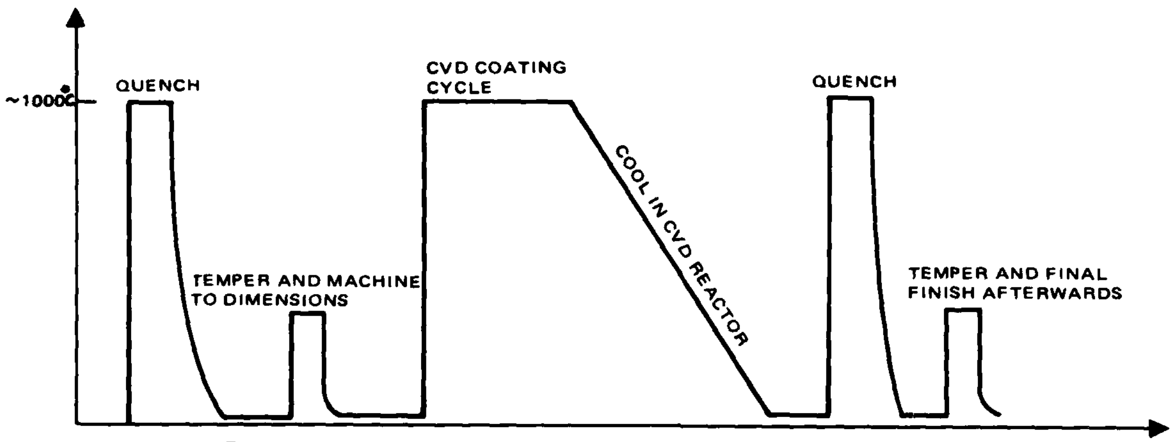

Figure 3. Schematic representation of the 440C bearing steel thermal history before, during and after the CrC-TiC CVD process.

(c) <u>The Present and Future of CVD Hardcoats for Rolling Element Bearings</u> - Based on the results of both European trips, we are convinced that the utilization of Swiss CVD technology and instrument bearing technology, currently using the CVD hardcoats for extending fluid lubricated gyro bearing wear lives, lends itself perfectly to our feasibility study. We are able to order hardcoated balls and races from selected Swiss firms now, coat them with solid lubricants in the U. S. and conduct the gyro study at Litton.

In fact, the first Hughes purchase order to LSRH, Neuchatel, Switzerland has already been placed, ordering the gyro bearing balls Litton needs for further work (see Table I). During our second trip, subsequent to a joint technical meeting between LSRH, RMB, Litton, Litef and Hughes, a preliminary specification was also prepared for CVD coated R-3, 440C ball bearing races (see Appendix A). LSRH and RMB are in the process of perusing this specification for the preparation of a price quotation to Hughes.

TABLE I. 440C STEEL AND CEMENTED CARBIDE GYRO BEARING BALLS (BARE AND HARDCOATED), ORDERED FROM LSRH/RMB/SAPHIRWERK COOPERATING SWISS FIRMS

BALL MATERIAL	COATING	DIAMETER, (NOM.) MM.	QUANTITY, EA.
440C (Double vacuum melted)	None	2.371	300
	None	2.376	300
	None	2.381	300
	None	2.386	300
	None	2.391	300
	CrC-TiC	2.381	300
WC + 6% Co. binder	TiC(1)	2.376	300
	TiC(1)	2.381	300
	TiC(1)	2.386	300
TiC + Ni-Mo binder(2)	TiC-TiN(3)	2.376	300
	TiC-TiN(3)	2.381	300
	TiC-TiN(3)	2.386	300

(1) Conventional TiC CVD, no CrC interlayer.

(2) Kennametal Kentanium Grade 162B or 165.

(3) Conventional TiC-TiN CVD, no CrC interlayer.

In spite of the overall advantages, our initial examination of the CrC-TiC coating revealed certain shortcomings of the process which the program must address to increase the chances of success for both gyro and turbine engine bearing operations:

(1) The CrC interface layer is rather uneven in the as-applied condition. After the re-heat treatment operation, diffusion evens out this layer to some extent. Nevertheless, the still uneven chromium diffusion into the TiC layer, probably caused by improper machining, followed by uneven acid etch, further causes differences in the hardness gradient throughout the coating at various radii (circumferential as well as cross-curvature) of the races or the balls. Chromium rich portions of the solid solution are always softer than the chromium-poor (i.e., titanium rich) regions, because TiC is significantly harder than the various stoichiometry version of the carbides of chromium. While the exact effect of uneven hardness of the top surface of the coating (after lapping the CrC-TiC layers to gyro bearing dimensions) on solid lubricant behavior is yet unknown, efforts should be expended for more even CVD of the initial CrC layer.

(2) Variations of surface hardness, as well as the total thickness of the lapped hardcoats on the balls and the races are aggravated by the fact that the hardness and dimensions of the bearing steel substrate are highly dependent on the CVD reaction temperature. As shown by Professor Ruppert's excellent paper from the Fachhochschule Wurzburg-Sweinfurt (see References 2 and 4), the anisotropic behavior of steels may cause problems in dimensional stability of the parts and in the distribution of internal stresses in the hardcoats. Because of the fact that CVD of chromium and titanium carbides is a high temperature process (i.e., requires temperatures exceeding 850°C, see Figure 3), the above problems are magnified. The underlying cause of anisotropic behavior, of course, is the preferred orientation of carbides in steels. For example, wrought material of as-quenched steels generally has larger dimensional changes

parallel to the direction of the carbide orientation than perpendicular to it. Pieces that have been quenched after austenitizing at 1000°C (the present CVD temperature) do not undergo a dimensional change in the perpendicular direction; the change in the parallel direction is, however, as much as 1 μm/mm.

These dimensional changes in small (e.g., R-3 size) gyro bearings can still be brought into line by lapping after coating. On the other hand, compensatory finishing can cause uneven hardcoat thickness around the bearings race (and ball) circumference and cross-race curvature. In the case of a larger bearing, such as the Type II specimen of the present program, this dimensional change can be disastrous, if certain remedies are not employed. One obvious remedy is to pursue the manufacturing of isotropic steels through rapid solidification [(e.g., the spinning disc method, or a magnetohydrodynamic approach to atomizing molten metal to produce fine, amorphous or microcrystalline metal powders (see Reference 5)], and use such steels for machining the M-50 turbine engine bearing speciments. Other, additional solutions are, in accordance with Professor Ruppert's recommendations, to tailor special pretempering, quenching, cold treatment and tempering cycles to minimize whatever distortion is inherent in any one heat of a bearing stock.

(3) Due to the fact that steel bearing balls are fabricated from wrought wire stock, CVD coating of precision steel balls is even more difficult to achieve with good coating thickness, hardness and dimensional control. This is one of the reasons why our program favors the use of coated cemented carbide balls (see Table I). The other reasons are higher load-carrying capacity and variable modulus over steel, lower specific gravity than steel (e.g., Ni-Mo bonded TiC) and CVD coatability using conventional deposition methods.

(4) The lapped finish of the CrC-TiC hardcoat on the RMB bearing races exhibits a pitted appearance. SEM photomicrography by Litton seemed to show that these pits are shallow. While these pits may not be harmful (in fact, may even be beneficial in terms of solid lubricant anchoring), pitting (and possible cracks) can be reduced in the future by a total CrC-TiC coating thickness of less than ~5 μm (the present approximate lapped thickness on the gyro bearing races). Additional removal of the TiC rich outer layer would expose the more CrC rich portions of the solid solution, which could be machined to a better finish. This latter step, of course, is largely dependent on even deposition of CrC, as previously discussed herein, and the effect of the hardcoat ductility change on the behavior of a solid lubricated gyro bearing.

(5) Among other factors, the stoichiometry of the carbides that form in bearing steel matrices are functions of the percent carbon and percent chromium in the steel. These diagrams expressing this quantitatively are available (Reference 6). Therefore, CVD of CrC-TiC to various steels requires special development of a compatible coating process for each composition. To date, LSRH perfected coating parameters for 440C only. Yet, Litton has most experience with the fluid lubricated version of 52100 R-3 bearings, because 52100 is the most popular bearing steel for gyro (and some other) applications. In view of the fact that bare 440C bearings are not available to the program due to long lead times, high cost, as well as pro and con arguments with respect to the relative desirability of 440C as a gyro bearing steel, direct comparison of bare and hard-coated 440C bearings will not be possible. Since Litton's research will focus mainly on the respective hardness value of the bearing balls versus the races and the location of the softcoat as to that of the relative hardness of the substrate, 52100 bearings can be substituted for the unavailable 440C ones. This substitution is acceptable, provided that no undesirable surface reactions or interactions will occur between the solid lubricant and the less corrosion resistant 52100 steel. This subject will be further discussed in the following paragraphs.

(6) Despite the Beilby layer-removing effect of the inherent HCl etching cycle of the CVD process and some metallurgical improvement of the machined bearing surface during the coating and re-heat treatment cycle, there is still some degassing between the CrC-TiC coating and the 440C interface. Degassing indicates improper cleaning, i.e., the liberation of trapped cleaning fluid vapors from the Beilby smear. This has been shown by Litton SEM photomicrography of cross-sectioned races. Therefore, the need for a thorough machining program still exists.

Coating processes considered second best at this time (e.g., reactive sputtering with TiN or plasma assisted CVD of other hardcoats) are lower substrate temperature processes and therefore do not provide the bearing surface alteration and diffusion needed for maximum hardcoat adhesion. These processes apparently need smear-free machined bearing surfaces far more than the conventional CVD process demands. The lower temperatures, of course, tend to alleviate over-tempering during those coating processes and the necessary retempering, where factors (e.g., retained austenite) may not be as well controllable as they are in any one heat of the basic bearing steel stock. This unfortunate dichotomy between these high and low temperature processes can be solved by other, more advanced methods. Litton scientists will evaluate other, improved coating processes and will make recommendations in Reference 2 as to the future path of hardcoating.

(d) <u>Layer-Lattice and Other Low Shear Strength Solid Lubricant Layers</u> - The hardcoat provides an ideal underlay for a low shear strength, solid lubricant top coat. Note that if this softcoat is very thin, the contact area is determined primarily by the yield pressure of the hardcoated steel substrate, while the force required to shear the junctions is determined primarily by the soft film. In the limiting case, as the film thickness approaches zero, in the equation

$$\mu = \frac{S_f}{P_s} = \frac{\mu_f P_f}{P_s}$$

where

μ = coefficient of friction

S_f = film shear strength

P_f = film yield pressure

P_s = substrate yield pressure

S becomes the bulk shear strength of the film material and P the yield pressure of the substrate (assuming S is independent of pressure, which doesn't always hold). It follows that μ for a low shear strength softcoat is equal to the product of μ for the film material in bulk form and the ratio of the hardness of the film material to substrate, as shown above.

Most layer lattice solid lubricants exhibit a more or less constant shear strength value above a certain critical load range. It follows that after a proper run-in (i.e., compaction) cycle, the consolidated solid lubricant film should exhibit a reasonably constant friction value in low (150 Kpsi) and high (250 Kpsi) load rolling element bearing application. As it will be shown by a forthcoming discussion, we may not need to vary the softcoat thickness during the feasibility (gyro bearing) study due to the above and additional arguments, as supported by known facts.

In addition to reducing the coefficient of friction and, therefore, controlling the traction of a solid lubricated bearing, a properly applied softcoat can act as a peak Hertzian stress-reducing entity (i.e., a cushion). Lower maximum bearing loads are synonymous with a lesser tendency for subsurface crack formation and crack propagation. This is important, especially in light of any adhesion-reducing effects the remnants of the partially healed Beilby layer would have on the CVD hardcoat, or in light of hardcoat processes less adhesion-effective than CVD.

Most of the initial Litton research will be conducted with RF or DC magnetron sputtered MoS_2 deposited by Hughes and other subcontractors, for two reasons:

(1) Exhaustive research during the past decades could not uncover an all-around, layer lattice solid lubricant better than MoS_2. While other dichalcogenides (e.g., $MoSe_2$, WS_2) are occasionally acceptable substitutes due to their better electrical conductivity or higher oxidative stability, none exhibit lower friction and longer wear life than MoS_2 layers.

(2) The theory and practice of MoS_2 sputtering is the most controlled among the advanced solid lubricant deposition techniques now available. There are several use histories of MoS_2-sputtered ball bearings with high reliability of operation.

As previously mentioned by Professor Cohen of MIT, there is a "surprising lack of knowledge concerning the stress-strain relation and glide systems of MoS_2 as a function of its substructure and 'alloying' ". We agreed with him in Reference 1, and research will be expended along those lines. Nevertheless, there is enough information available on sputtered MoS_2 and surface roughness-lubricant film interaction to predetermine the bounds of the film thickness for deposition on gyro bearing surfaces.

For example, previous work by this writer (Reference 7) has shown that one must be careful in correlating CLA and RMS surface roughness ratings with sputtered MoS_2 substrate (or any lubricant substrate) behavior. A variety of basic forms in surface waviness could yield identical CLA/RMS ratings and still exhibit quite different tribological performance characteristics (e.g., lubricant retention and anchoring). The heights of the tallest bearing steel roughness peaks, as measured

from the reference plane, as well as their frequency and distribution will influence the number of sliding or rolling cycles to failure. (The stoichiometry and crystal structure of the MoS_2 layer is assumed to be ideal). It was found that stylus traces of standard friction and wear test specimen surfaces indicated peak heights of two to four times the CLA (or AA) reading. This range agrees with previous findings by others that the RMS or CLA (AA) values are usually 1/3 to 1/4 of the total depth.

Since the extent of metal-to-metal or, in the present case, metal-to-hardcoat or hardcoat-to-hardcoat contact is the function of MoS_2 removal from the highest surface roughness peaks, on first order the film thickness should be slightly more than two to four times the CLA (AA) readings. Since the bare and CVD hardcoated bearings surfaces will be around 1 μin CLA (discounting the effects of occasional CrC-TiC surface pits), the first order bounds of the MoS_2 film thickness are from 500 to about 1000 Å. Second order film thickness estimations will be offered by forthcoming computer work, performed separately by Litton and by MTI/SKF, and will be checked by Litton gyro bearing tests.

While MoS_2 appears to be an ideal lubricant to prove the feasibility of solid lubed gyro bearings, it does have three major shortcomings which would influence its effective behavior at high temperature operation in air (on the Type II bearing), or under high humidity conditions even at ambient temperatures:

(1) Above 600°F, its oxidation rate is excessively high.

(2) In high humidity, its friction increases significantly.

(3) MoS_2 has the tendency to promote corrosion of steel in high humidity environments.

With respect to Point (3), in view of the anticipated use of bare steel 52100 gyro bearings in conjunction with MoS_2, the effect of ambient laboratory air and humidity on the corrosion mechanism of the ball bearing surfaces appears to be a prime object of interest. Concern stems from the fact that molybdenum disulfide promotes the corrosion of metallic surfaces by reacting with oxygen and water to form molybdenum trioxide and sulfuric acid, as described by the following equation:

$$2MoS_2 + 9O_2 + 4H_2O = 2MoO_3 + 4H_2SO_4.$$

The extent of this reaction is controlled by the reaction rate, which, in turn, is a function of the activation energy, the rate constant, the humidity level, and the temperature. The magnitude of the activation energy itself is, to some extent, temperature dependent. Since the time dependence, the limits of temperature and relative humidity at which corrosion of MoS_2-coated semi-corrosion resistant 440C steel or even more marginally corrosion resistant 52100 will occur are not precisely known, Litton intends to alleviate this potential problem by close control of the handling and processing environmental humidity. Their past research indicates that corrosion of 52100 is a function of a critical relative humidity (similar findings have been known for years) and the severity of corrosion can vary from geographic region to region within the U.S. The West Coast of California appears to be one of the milder areas with respect to corrosion tendencies. These data, along with research by this writer, showing that MoS_2 burnished 440C will not exhibit any corrosion after twenty-four hours in condensing humidity, lends credence to Litton's intentions. Indeed, if moisture is eliminated from the environment, the products of the MoS_2 oxidation are MoO_3 (a solid) and SO_2 (a gas), as described by the equation

$$2MoS_2 + 7O_2 = 2MoO_3 + 4SO_2$$

Discounting any surface sorption and assuming an open system, the gas should be free to leave the site of oxidation.

Of course, the temperature usability limit of MoS_2 under dynamic condition depends on the reaction rate of MoO_3 formation and the simultaneous removal of the oxide from the parent material under sliding rolling conditions.

The formation of MoO_3 is a second order reaction. A first order reaction implies that the oxide layer is not protective and does not interfere with continued reaction. Molybdenum trioxide does form a protective oxide layer and further reaction is limited by the rate at which oxygen diffuses through the oxide surface coating and by the rate of surface oxide removal by tribological action.

With respect to frictional behavior, it is well-known that the friction of hexagonal (normal lubricant grade) molybdenum disulfide is highly dependent on the relative humidity. At high humidities, the coefficient of friction is high, at low humidities, it is low. The phenomenon is attributed to "hydrogen bonding" of the flat-platelet MoS_2 crystallite edges which prevents their free "card-shuffle"-like displacement relative to each other. Desorption of the entrained moisture either by static or by frictional heating or by vacuum degassing brings about the desired reduction of friction. Although the friction of MoO_3 is higher than that of the underlying MoS_2, it is by no means excessively abrasive, nor is it difficult to remove during sliding. Although it is commonly believed that MoS_2 is hygroscopic, the belief is not valid because the non-oxidized MoS_2 surface is in fact hydrophobic. It is the oxidized surface itself which tends to be hydrophilic. Since commercial MoS_2 powder is invariably exposed to air prior to use, each particle (on a sputtered film, to a lesser extent) is covered with a thin oxide layer, imparting slightly hygroscopic qualities to the powder or the film surface.

Since the long term gyro bearing tests will be run in a reduced pressure of dry hydrogen gas and the BET surface area of powders is far higher than that of a compact sputtered film, oxidative wear and high

friction of this solid lubricant during any short term gyro bearing tests in ambient air should not be objects of concern.

In order to be responsive to the higher temperature (500°F) requirements of the Type II feasibility study bearing specimen, MoS_2 appears to be marginally useful. For the higher temperature limits, MoS_2 needs to be modified, or other solid lubricant systems must be used. Forthcoming Litton bearing tests may also reveal better solid lubricant candidates for gyros than MoS_2 (see Reference 2).

(e) Gyro Bearing Retainer Preparation - Due to the characteristic, uneven transfer film layers deposited by polymeric self-lubricating composite retainers, the gyro retainers will exhibit the required high specific strength through the use of polymeric, but not necessarily self-lubricating composites. Lubrication will be provided by the same sputtered softcoats that will be applied to the balls and/or races.

In its benign role, a retainer (AKA ball separator or cage) will not be a source of lubrication, but neither should it be a source of irritation. The most important points here are:

(1) The geometric design of the cage;

(2) The friction/wear reducing role of the sputtered lubricant in the plastic composite ball pockets and on the land areas (where the cage rubs against the races); and

(3) The physical constants of the cage hoop, e.g., dynamic modulus, hardness and spring constant, dimensional stability, homogeneity and hoop strength.

We will lean heavily on computer programs to help refine parameters to the optimal. At this time, however, we don't know the

exact dynamic modulus, tensile/compressive strength, and other requirements for the ideal retainer material to be used for gyro application. Therefore, based on the previous composites research performed at Hughes, we will provide various materials with a range of property combinations and check computer predictions with bearing tests at Litton. The conventional cotton-reinforced phenolic will be used as the baseline cage material.

It should be noted here again that during the initial (feasibility study) portion of the program, neither the gyro nor the turbine engine test temperature will exceed 500°F. Consequently, the Hughes-developed high temperature (600°F) and high load-carrying (25 Kpsi) capacity, Thermid 600-based self-lubricating composites can serve as model compounds for new, more advanced materials, to be used as stock for ball bearing retainers for both types of bearings.

The attached test program plan (see Appendix B) describes the first half of our in-house composite formulation efforts, i.e., the preparation and testing of composite candidates that appear likely as gyro bearing retainer materials.

The second half of our efforts will consist of impregnation and testing of special, 3D cylindrical weaves of carbon, graphite and carbon/graphite hybrid weaves, to be prepared by Fiber Materials, Incorporated (FMI), in Biddeford, Maine (see forthcoming discussion). These materials will comprise the turbine engine retainer stocks. The estimated time frame of the latter portion is an eighteen month effort, starting after the completion of the gyro bearing retainer work.

(f) Computer Modeling Plans - We will lean heavily on Dr. Gupta's modified DREB program (by MTI) for the necessary predictive gyro and turbine bearing diagnostics, pointing the way to both cage design and frictional as well as mechanical cage requirements for retainer stability at various loads, speeds and general bearing designs (see References 8 through 11).

Other rolling element bearing computer programs which can be complemental or supplemental to DREB are also planned for incorporation, as explained below.

(1) DREB is the only truly generalized dynamic formulation of bearing behavior, emphasizing the general, six degrees of freedom motion of the ball, the ball/race and race/cage interactions, drag losses in ball bearings and other factors. It looks at a bearing from a systems viewpoint, but isolates the bearing in space as a "mini-system" with respect to an inertial frame of reference. In real life, however, each bearing itself is a part of a larger system, where the behavior of that bearing is only a nodal (i.e., partial) input. The input is not only kinematic, but also thermal in nature, be it a high temperature, single-bearing tester or an entire gyro platform or turbine engine containing several bearings. Therefore, a unique system analysis capability expanded from the analysis of a single component appears essential for the investigation of advanced solid lubricated (or any lubricated) bearing system design concepts. The realistic evaluation of dynamic and thermal effects on load and motion support capability requires the assessment of the interaction between a shaft, the bearing or more than one bearing, and the hardware thermal environment. The basis for such predictability exists today at SKF Industries, Incorporated, King of Prussia, PA, in the form of a computer program named SHABERTH (SHAft BEaring THermal, see References 12 and 13). Although this program is only quasi-dynamic, its credibility has been examined in the light of experimental evidence and found to exhibit reasonably good correlation between theory and practice of fluid lubrication.

(2) SHABERTH can do double duty by defining a material property window through which material development scientists must pass their solid lubricant/retainer materials. The program must be capable of extensive system parameter exploration to achieve this. As a consequence, a system survival time parameter can emerge in the foreseeable future for a particular solid lubricant system within a particular hardware configuration. The most attractive part of this approach is the

fact that in our later research, such predictions will occur before rather than after the accomplished fact of a particular material selection or development. It is encouraging that SHABERTH, in spite of its quasi-dynamic nature, has seen extensive use over a period of years and is thus under analytical control. Therefore, modifications to accept increased flexibility (i.e., applicability to solid lubrication) will encounter minimum development surprises. SHABERTH will be able to give us general approximations before DREB will provide us with the final specific answers.

With respect to existing plans, both the DREB and SHABERTH programs will be first reworked to initially predict the behavior of a solid lubricated bearing not operating with a self-lubricating retainer (also our initial gyro concept). The retainer will only provide a benign, stable environment by appropriate cage design and a proper lubrication of the ball pockets and the land areas with the same incompressible solid lubricant of a given thickness and traction value found on the balls and races also. The traction values will be determined by some empirical tests of simple geometry or by the use of previous published data from reliable sources. This first step is both simple and reasonable, because under high Hertzian stresses, fluids do undergo a liquid-solid transition in the concentrated contact zone. This rheological change can yield a reversible alteration of the fluid into and out of an amorphous solid region between the balls and the races. Of course, both computer programs were originally designed for the prediction of fluid lubricated bearing behavior and it follows that where liquid lubrication leaves off, solid lubrication can begin. During this first step, only the behavior (e.g., retainer stability, torque, etc.) and not the wear life of the solid lubricated bearing will be determined.

Here, any wear life of the bearing will be determined only by bearing stability and the endurance of the solid lubricant layer. To give an example, a bearing like this would be represented by a gyro bearing specimen sputtered with MoS_2 on the balls or races, and on the retainer.

With respect to accomplishments to date, in December 1978, Dr. Gupta of MTI received a letter of recommendations from Mr. B.D. McConnell of AFML (our Project Engineer), for the use of the NYU computer facility for the above program. An application for the computer time has been filed. However, no response has yet been received. As soon as the computer time request is approved, the computer modeling task will be initiated.

SKF's involvement will depend on future program negotiations, in accordance with the plan described above.

Litton's gyro bearing test results will serve as initial tools of proving or modifying both DREB (and SHABERTH) for better predictability of solid lubricated bearing behavior as rapidly as possible.

(g) Gyro Bearing Test Plan

(1) Geometry Inspection - Nominal bearing geometry and variations play a major role in establishing bearing performance. With regard to life, geometry influences the stress levels and thickness of the liquid EHD film. It is anticipated that lubricant film thickness is of equal importance for solid lubricant systems. With regard to gyro performance, poor geometry influences bearing dynamics and causes unwanted vibration. It is known that the lubricant film is influnced by local stress levels, which in turn, are affected by ball path runout and cross-race curvature. Cross-race curvatures will be closely controlled, because thickness of the lubricant film in the pressure zone is a function of the local cross curvature. The IndiRon (shown in Figure 4) and other roundness-measuring devices will be used before and after application of the hardcoat/softcoat system to assure that geometry levels meet current requirements. There is special fixturing available for measuring cross-race curvature and there is a Proficorder for surface finish determinations for before/after coating measurements (see Figure 5).

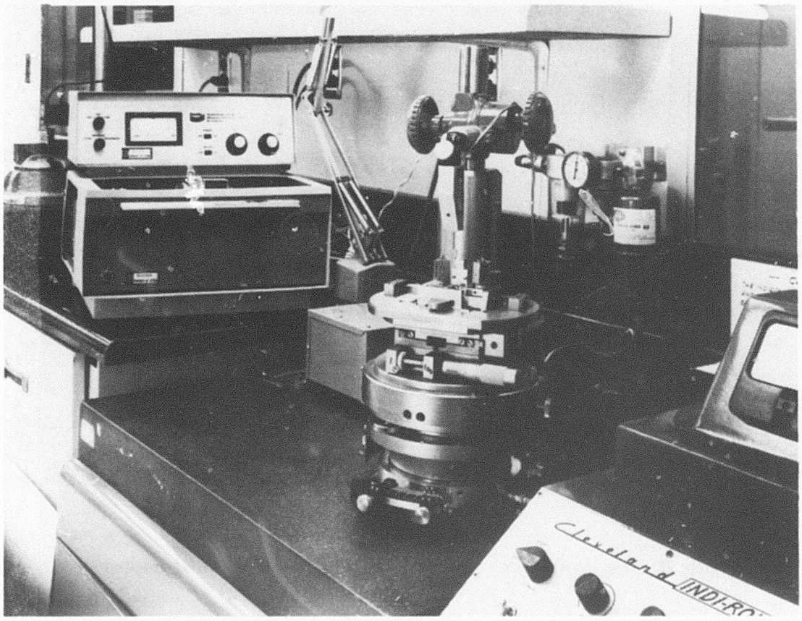

Figure 4. The IndiRon roundness measuring device.

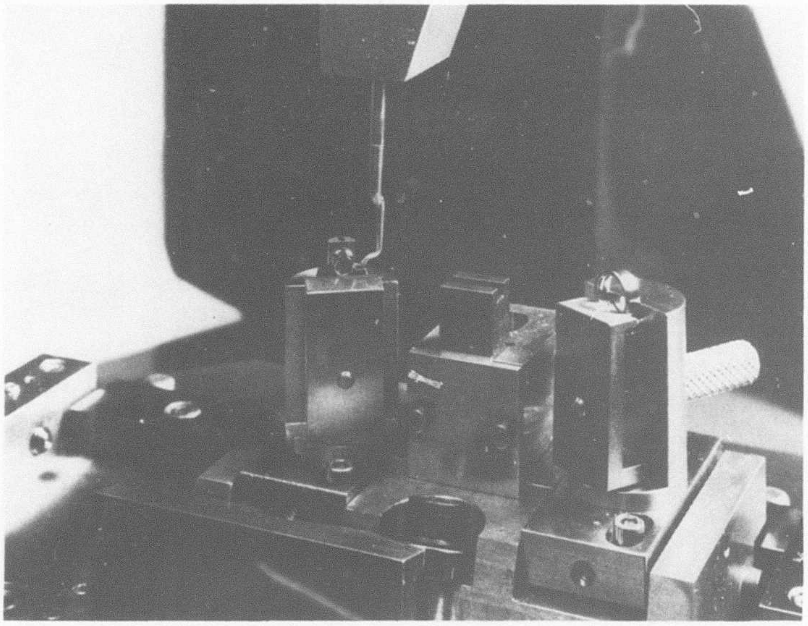

Figure 5. The Proficorder surface roughness measuring instrument.

(2) Preliminary Screening and Evaluation of R-3 Bearings on the Low/High Speed Tester - The Litton single bearing tester permits first low speed, then high speed testing of one bearing at a time. A moderately comprehensive test requires about four hours, permitting the examination of two bearings per shift. Testing of the most perfunctory sort requires about 1.5 hours. Combinations which perform poorly may be identified in as little as one hour.

The high speed tester (Figure 6) provides the loads and speeds the bearings see in service. As a first test, coating thickness variations, poor adhesion and rapid wear will be revealed. Retainer

stability and both average and dynamic variations of torque will be measured, including their vibration spectrum. The bearings will be observed stroboscopically under low magnification; ball speed measurements will permit further data inference.

(3) Coating and Life Testing of Bearings - From all of the preliminary coating evaluations and bearing screening tests, the most promising and/or interesting material combinations will be assembled in pairs into test fixtures accurately simulating the gyro environment. The test groups will be compared not only to the relative baseline of solid lubricated

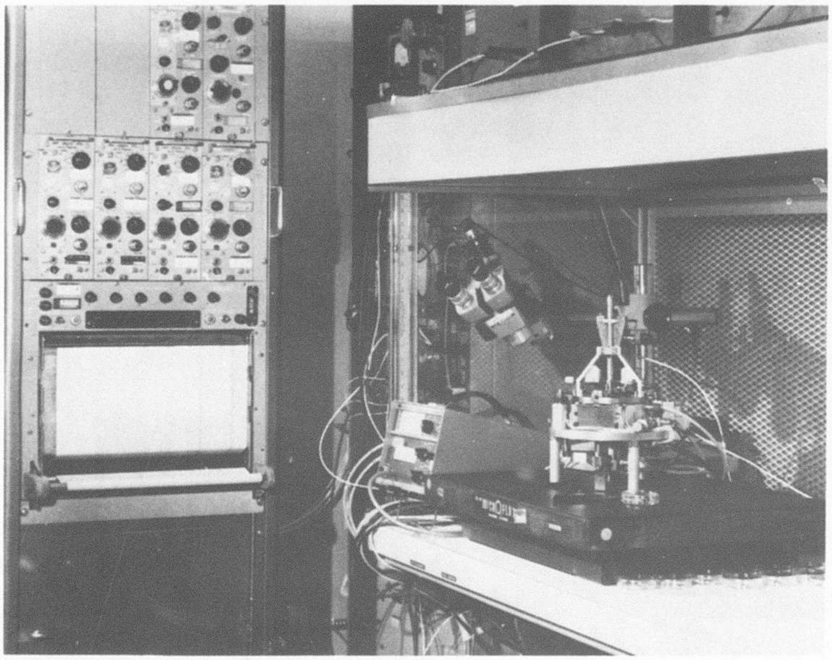

Figure 6. The Litton high speed gyro bearing tester.

bearings available today, but to the absolute baseline of a similar group of conventionally configured oil-lubricated bearings to provide a data base for the bearing lot and test condition influences.

Bearing operating conditions are monitored by an extremely sensitive differential wattmeter, a test widely used in the gyro industry and called a milliwatt test. Perturbations in the bearing function caused by wear, debris, lubricant loss or retainer instability are reflected in torque variations or torque increases. The torque changes will be observed and recorded as variations in power demand on the synchronous hysteresis-type gyro motors. For the typical life test setup, see Figure 7.

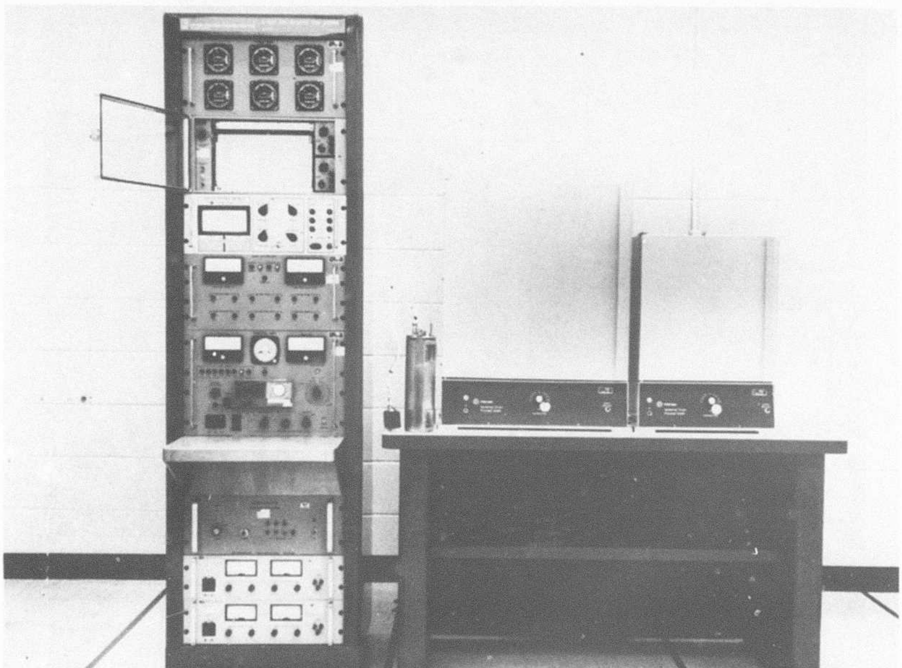

Figure 7. The Litton gyro bearing life test assembly.

At the outset, milliwatt measurements will be made immediately at startup, again at about 48 and 96 hours, and then at weekly intervals. After 60 days, the test period will be lengthened to monthly intervals for units which remain stable.

Throughout the test program, surface metallurgy and surface analytical examinations will be conducted in combination with the high speed and life tests for the identification of friction and wear mechanisms.

Final selection of the ideal hardcoat/softcoat system for gyro bearings will be determined after completion of all studies. Special attention will be paid to answering the computer technologist's questions through gyro bearing performance data, serving as the first confirmation or denial of the initial bearing dynamics programming efforts.

2. Turbine Engine (Type II) Bearing Research

The increase in the importance of the cruise missile turbine engine, solid lubricated bearing technology has placed this bearing in the focal point of all research as expained previously.

In order to be responsive to program requirements, we had to select a ball bearing test specimen representative of one used in advanced cruise missile turbine engines. At the same time, the bearing material had to satisfy the feasibility study 500°F temperature requirement. It also had to offer some promise of usefulness at the next higher temperature limit 1000°F.

After consultations with SKF, we selected a 1.18 in. bore, 1.85 in. OD, split inner race, AISI M-50 steel bearing (R_c 60 min.) from an advanced cruise missile turbine engine (Reference 16). This bearing is presently designed to operate in an oil lubricated condition employing a silver plated SAE 4340 steel, one piece retainer.

Note that previous research indicated that AISI M-50 steel produced the most favorable life results at elevated temperatures, when compared with other high-temperature bearing steels (Reference 17). In our program, the retainer will be replaced by specimens provided by Hughes Aircraft Company in cooperation with the eventual fabricator of the bearing. Note that Litton is expending considerable effort at this time to identify likely sources of turbine engine bearings machined to a smear-free surface condition. Mr. Baginski is examining specimens from U.S. firms, such as SKF, and Western European firms, such as FAG Kugelfischer in Schweinfurt, West Germany. We hope that the results of this investigation will point to a source of likely turbine bearing specimen source, without resorting to an extensive machining study first under separate sponsorship. Although it is generally accepted that the surface stresses induced by machining the larger turbine bearings are less than those of the smaller gyro bearings, the question of the final procurement source for the former has still not been answered. Again, it is hoped that plans for a separate machining study are immediately forthcoming elsewhere to aid in solving this prevalent problem.

As explained before, MTI is the major subcontractor on the Type II bearing, providing capability for both computer simulation of bearing dynamics and turbine engine bearing tests under the same roof. The only reasonable approach at this time is to start with computer modeling to predict likely modes of instability for both Type I and Type II bearings, followed by the incorporation of advanced materials available at the start of the experimental work.

(a) Computer Modeling Plans

The Dynamics of Rolling Element Bearings computer program, DREB, will provide the foundation for the computer modeling of the solid lubricated ball bearings in the project. Within certain approximations, the present form of DREB can be applied to solid lubricated bearings by specifying appropriate friction coefficients. For a much closer simulation,

it will be necessary to make certain modifications to the program. These required modifications can be readily identified by making a few preliminary runs in the light of some available experimental data on the general performance of solid lubricated ball bearings. The following tasks will schematically subdivide the total effort required for computerized simulations of solid lubricated ball bearing performance:

- Task I - Friction and Wear Experiments - The purpose of this task is to generate the friction and wear data that will be a fundamental input to the DREB computer program for simulating the performance of solid lubricated ball bearings. This task represents an extension to the scope of work originally designated by Hughes and should begin as soon as additional negotiations are over, in order to assure proper interfacing with the experimental program.

The pin and cylinder-type friction and wear tester presently existing at MTI will be modified to suit the operating conditions relevant to the solid lubricated ball bearings. The first modifications will consist of the installation of a furnance around the test specimens and the building of a heat dam on the drive shaft. This will make the desired friction and wear experiments at high temperatures possible. The second modification will involve the replacement of the pin with a suitable fixture which can hold a ball. Also, an additional fixture will be installed so that a specimen of the cage material can be held against the rotating cylinder. This will ensure a realistic transfer film at the ball/cylinder contact.

After the above modification, a series of friction and wear tests will be undertaken over the range of operating conditions for the two types of bearings under consideration. All test specimens will be purchased by MTI and supplied by the materials or bearing manufacturers participating in the present project. The total test sequence will be limited to a set of about twelve experiments for each material pair and a maximum of eight materials pairs may be tested.

All the data will be documented properly so that it can be readily used as input to the DREB computer program.

- <u>Task II - Computer Simulation of Type I Gyro Bearings</u> - As previously explained, performance simulation of the Type I gyro bearings and interfacing the analytical predictions with the experimental observations are the primary objectives of the initial portion of MTI's tasks.

MTI will interact with Litton and determine the appropriate geometry, materials and operating conditions relevant to the Type I gyro bearings. The information drived from Task I will be used to determine the appropriate traction slip relationships in the bearing and run the updated DREB computer program to obtain some preliminary simulations.

The experimental setup at Litton will be reviewed and DREB will be used to obtain a number of simulations corresponding the extreme operating conditions. This will be done in parallel with the testing at Litton. The analytical torque predictions will be evaluated in the light of the experimental data and DREB will be modified further if necessary.

- <u>Task III - Other Preliminary Runs on Existing DREB</u> - For the typical Type I and Type II bearings and prescribed operating conditions, a few simulations will be obtained by using the present form of DREB. The purpose of these runs will be first to "scale up" DREB for the specific application under consideration and, more importantly, to review DREB capabilities in the light of existing experimental data on solid lubricated ball bearings and the general design requirements set for the present project. This will basically result in a critical review of the desired geometrical, materials and operational parameters and some initial recommendations for the required updates to the present form of DREB. It is expected that some of the modifications will be concerned with ball/cage pocket wear, changes in contact stresses due to coated races, thermal distortion of the various

bearing elements at elevated temperature and appropriate traction slip relationships for the solid lubricants under the prescribed operating conditions.

- Task IV - DREB Modifications - The refined modifications identified in the above task will be analytically formulated and appropriate computer codes will be developed for these added features. These new codes will then be incorporated in DREB and a number of test cases will be executed in order to fully debug the updated version of DREB.

- Task V - Pretest Runs on DREB - Prior to the undertaking of the high load and high speed bearing tests, a few runs will be made on DREB and the preliminary designs will be evaluated. Some design changes or recommendations may be made as a result of this evaluation. Also, a few parametric runs to cover the anticipated operating range will be made and the predicted bearing behavior will be studied.

In addition to the above tasks, it should be pointed out that a strong interaction between MTI and other participating organizations is essential in order to bring the total program to fruition. All the experimental work must proceed with close ties to the analytical development and the computer simulations.

It should be reiterated that in the case of the Type I bearing, all critical bearing surfaces are coated with thin, sacrificial films of the softcoat (retainer included) and once the coating wears off, the bearing will probably have failed. The retainer here, therefore, is not a self-lubricating entity.

However, in the case of the Type II bearing, we must consider not only the above model but one employing the self-lubrication concept also.

This second model will consist of the consideration of the self-lubricated bearing through examination of the lubricative cage. Under this condition, an ideal ball/self-lubricating retainer ball pocket interaction would be represented by controlled transfer of the composite retainer material by the rubbing action of the ball. The above wear action should form and maintain an evenly thin, but wear-preventing film on the balls and the races. In view of the lubricant source, the computer program should be able to predict not only the torque increase due to excessive wear of the ball pockets and increase in the transfer film thickness (i.e., reduction in bearing clearances), but changes in the dynamic stability of the retainer due to continuous variations in its moment of inertia and cage imbalance due to wear. This variation comes about from changes in ball speeds from ball to ball and the resultant changes in ball to ball pocket loads. Composite retainer ball pocket wear rate is, of course, directly proportional to those loads. The second generation of computer forecasting will be used both for predicting instantaneous bearing behavior (e.g., cage instability) and total bearing wear life.

In our opinion, achieving the above goals by the two suggested computer approaches is not only theoretically sound, but also cost effective, because we are not depending on a single set of ideas. Thus, we will increase the chances of success by defining physical data needed to enhance credibility of simulation results and recommending experimental programs to acquire such data.

Of course, the combination of the above two approaches is also possible, especially for extremely demanding applications (e.g., operation at 1000°F and above). SKF's computer studies will supplement MTI's, along the same path delineated above.

(b) High Temperature Hardcoat/Softcoat Combinations and Self-Lubricating Retainer Materials

As explained before, the hardcoat technology for Type II bearings is still undeveloped and requires further, extensive study.

With respect to high temperature, low shear strength solid lubricants for softcoat applications, MoS_2 needs to be modified or other solid lubricant system need to be used:

(1) MoS_2 can be blended with an oxygen scavenger, which will not harm the frictional quality of the solid lubricant. It is known that there is a synergistic interaction between MoS_2 and Sb_2O_3, since the latter preferably oxidizes in mixtures of the two. This reduces the oxidation rate of MoS_2 to MoO_3, lowering the friction and increasing the wear life of the composite layer. Note that TMI, Santa Barbara (one of our potential subcontractors) has developed sputter deposition of MoS_2-Sb_2O_3 films for solid lubrication of gas bearing surfaces. Thin films of this lubricant would be more conducive to sufficiently long turbine bearing wear lives in bearing tests needed in correlating theory with practice. If the synergistic mechanism hypothesis is correct, antimony in its elemental form should be an even more effective oxygen getter.

(2) "Alloying" of MoS_2 has been effectively accomplished by the research firm Hydromecanique et Frottement (HEF), St. Etienne, France, in cooperation with local universities. By the development of the Fe/Mo/S ternary phase diagram (see Figure 8, taken from Reference 19), they have discovered that the composition bounded by the

$$Fe_{0.66}Mo_3S_4 - FeMo_5S_6 - FeMo_3S_4$$

triangle yielded a solid lubricant material that exhibited better friction and wear characteristics than MoS_2, both in air, in inert gas and in vacuum

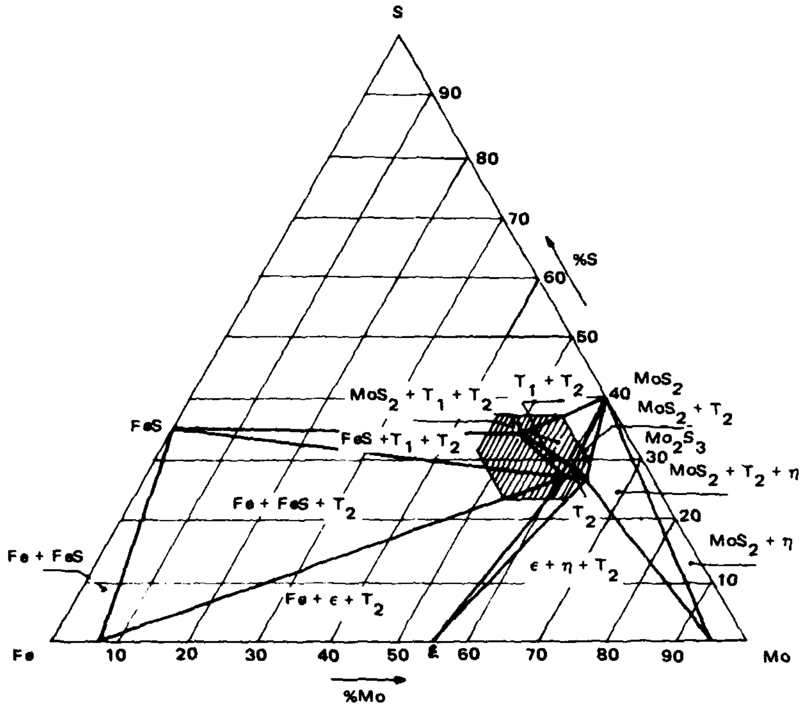

Figure 8. The ternary phase diagram of iron, molybdenum and sulfur.

(Reference 20). This material can be produced both in a self-lubricating compact form or can be magnetron sputtered in HEF's large, Leybold Heraeus ion plating/sputtering equipment combination (Figure 9). While the additional research connecting crystal structure, tribological behavior and oxidation stability has yet to be performed, the base material will presumably be available for investigation. We are currently planning to purchase a sputtering target and sample slugs of the materials for our own use. The cooperative attitude of HEF offers considerable encouragement for further work.

(3) Dr. Warren Jamison of the Colorado School of Mines (also the chief executive of TMD, Evergreen, Colorado) is one of the few members of academia who has done fundamental research on the behavior of solid lubricating materials. His previous research along the lines of advanced solid lubricants has prompted Hughes to invite him to participate

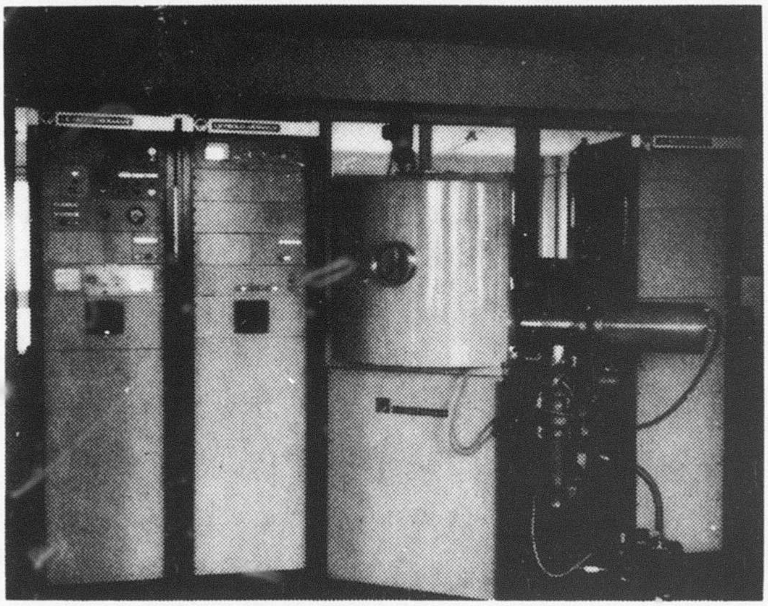

Figure 9. HEF ion plating and magnetron sputtering combination equipment.

in the present program. In his subsequent technical discussion forwarded to Hughes, Professor Jamison aptly stated that "molybdenum disulfide has been well established as a commercial solid lubricant for several decades. Its major applications as an additive for grease and as a pigment in bonded solid film lubricants has somewhat masked the fundamental mechanisms which promote its lubrication effectiveness. The observations that MoS_2 has a layered crystal structure with low shear strength between layers, and that the layers became oriented under sliding conditions with their planes of easy shear parallel to the motion has been accepted as an explanation of the inherent lubricity of this material. However, as Magie (Reference 19) and Jamison (References 20 and 21) have shown, MoS_2 belongs to a family of layered materials with similar crystal structure and with various frictional properties ranging from abrasiveness to excellent lubrication.

Jamison showed that both the lubrication effectiveness and structure of these materials depends on the number and distribution of electrons within the material. He also showed that both the

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structure and good lubricating properties of MoS_2 can be synthesized, using materials which are poor lubricants, by controlling the electron concentration. The synthetic lubricants were formed by combining metal atoms with the appropriate numbers of chalcogen atoms in various ratios to effect the required electron concentrations. Whereas this technique was satisfactory to demonstrate the validity of the theory, it is inappropriate for the development of good, stable solid lubricants because the materials are relatively scarce and no effort was expended to ensure thermal and hydrolytic stability or to optimize other properties which are desirable in operative systems.

Jamison (Reference 22 and 23) has suggested methods for creating stable solid lubricant materials by selecting structures with desirable secondary properties and then adjusting the electron concentration to provide the good lubricating properties. One method by which this should be possible is substituting a small portion of the metal atoms comprising the structure with other metal atoms, such as in creating electronic semiconductors. Another method for adjusting electron concentration without altering the basic structure is to intercalate small electron donor atoms into the spaces between the layers of the structure (Reference 23). The purpose of the research proposed herein is to explore the preparation techniques and effectiveness of synthesizing solid lubricant materials according to the principles outlined by Jamison (References 20 through 23)."

We intend to cooperate with Professor Jamison on research conducted to further the understanding of how solid lubricant materials may be synthesized to provide lubrication in the manner of MoS_2, while also providing superior secondary properties (thermal stability, hydrolytic stability, etc.).

"Two systems will be studied: chromium sulfides and titanium disulfide. The first system, CrS_x, would have the same electron concentration as molybdenum disulfide with $x = 2$ (CrS_2). However, chromium sulfides have a wide range of compositions on either

side of CrS_x (as opposed to MoS_2, which is not observed to vary appreciably in its stoichiometry). In addition, CrS_x avoids the CrS_2 composition. The structure of the system CrS_x has been well documented by Kjekshus and Pearson (Reference 24) and others (References 25 and 26), although no fundamental differences from MoS_2 have been disclosed. It is believed that by adjusting the electron composition, by substitutional or interstitial doping, and possibly by adjusting the average atom size ratio by replacing a certain fraction of the sulfur atoms with larger selenium or tellurium atoms, the structure and lubricating properties of MoS_2 may be effected.

The second system, TiS_2, has a rigid stoichiometry and exists in the classic $Cd(OH)_2$ type structure, which is characteristic of compounds of this type with two less electrons than MoS_2. Attempts will be made to stabilize the MoS_2 structure and properties by doping TiS_2 to increase its electron concentration.

The material preparation procedures which will be used are those developed previously (References 20 and 21), which consist of direct reaction of the elements in evacuated vials at high temperature. Should this technique prove difficult for some solid solutions, iodine transport reactions have been developed (Reference 27) which are simple and effective. The intercalation of interstitial doping compounds in TiS_2 has been thoroughly developed and documented (Reference 28).

Structural parameters of the sample materials will be determined by X-ray diffraction analysis. Although TMD does not possess scientific X-ray analytical equipment, leased time is available on a new Norelco diffractometer at the nearby Colorado School of Mines.

Screening tests of lubricating effectiveness will be conducted on a Timken-type tester at TMD. The hydrolytic and oxidative stability test can be conducted by a commercial manufacturer* of solid

*E/M Lubricants, Inc., West Lafayette, Indiana.

lubricants with whom TMD has a working agreement. Samples will be made available to Hughes for evaluation.

Jamison and Weber showed (Reference 29) that lubricating effectiveness of solid lubricant materials can be correlated with contact potential measurements. Jamison has briefly described the theory behind this effect (Reference 23). The measurements, while simple to make, are very powerful in confirming the theory of lubrication with lamellar solids. However, meaningful data must be collected under high vacuum conditions. TMD does not currently possess such equipment." It is planned that Hughes equipment will be provided on loan or a rental basis, so contact potential measurements can be made a part of the materials research.

"It is expected that the research to be performed by TMD will advance the state of knowledge of the mechanisms by which MoS_2 type compounds provide effective lubrication. In addition, knowledge will be gained which will advance the state of the art in preparation of synthetic solid lubricants which have superior lubrication and thermal, oxidative and hydrolytic stability to that of MoS_2."

(4) Along the lines of intercalates, graphite has been known to form compounds by the inclusion of foreign atoms within its structure. Fluorine, one of the most reactive elements, reacts with graphite without combustion from about 790°F to 1022°F, forming a grey-colored solid. The approximate composition of this solid is $(CF_x)_n$, where x can range from 0.7 to 1.1. The most prevalent form is $(CF_{1.1})_n$. In some low-load tests, this solid has been shown to be a promising high temperature pigment in high temperature resin binders and composite platings. More recently, considerable attention was given to the fundamentals of its lubricating activity, some discrepancies in its reported usefulness and the practicality of its use in aerospace hardware. The superior thermal-oxidative stability of graphite fluoride (see Kamarchik's and Margrave's latest works, References 30 and 31) suggests its use as a sputtered layer (never before attempted), if its load-carrying capacity proves to be

sufficiently high. The incorporation of the $(CF_x)_n$ may preclude the need for inclusion of an anti-oxidant. A thorough literature survey of this subject may be found in Reference 32.

(5) Boes and Chamberlain (References 33 and 34) intermixed WSe_2 (-200 - 325 mesh powder) with a gallium-indium alloy in the liquid state. The mechanism of interaction between the constituents is such that 5 to 25 weight percent of the metal reacted with the solid lubricant. The powdered lubricant/metal aggregate was compacted in a die at room temperature and high pressure, followed by a carefully controlled sequence of high temperature curing at progressively increased temperatures up to 566°C (1050°F). The resultant material was medium hard (Scleroscope hardness = 37), machinable and drillable and resists oxidizing environments at temperatures two to three times higher than the parent lubricant, WSe_2. The composition (AKA the Westinghouse compact) in weight percent was 90 WSe_2/10 GaIn, where the eutectic itself was 80 Ga - 20 In. This material was shown to be an effective wide temperature range bearing retainer as pressed-in inserts in metallic cage shrouds (Reference 35), also exhibiting good friction and film transfer characteristics in Hughes and other industry friction tests at laboratory ambient temperatures. Gardos and Castillo (Reference 32) also demonstrated that the Westinghouse compact, in the pulverized form, constituted the most thermally stable and tribologically effective additive in their 3D carbon reinforced, high temperature — high load, self-lubricating composites based on a polyimide resin.

It is unfortunate, however, that aside from engineering friction and wear tests, little effort has been expanded to date on identifying the compact's fundamentals, both with respect to physical parameters (crystal structure and the resulting melting point, strength, friction and wear rate) and chemical parameters (stoichiometry, corrosion preventive capacity at high temperatures and oxidation resistance).

In the present work, we have already begun preparations for producing a sputtering target from the Westinghouse

compact. Since target fabrication is a specialized process, we are purchasing the base mixture from Westinghouse and will have the target pressed, cured and mounted to a backing plate at the Materials Research Corporation (MRC). The latter is a specialty house dealing with the preparation of vacuum deposition equipment and materials.

Also, to the best of our knowledge, Westinghouse is now conducting in-house research to identify the links between the practice and theory of their compact. Based on successful demonstration of their ability to identify the stoichiometry, crystal structure, the nature of exothermic reactions during cure and other fundamental parameters, their further involvement would constitute the preparation of an improved self-lubricating compact in FY'80. Note that Mr. Baginski of Litton has also conducted some identification work on this material and will present some data in Reference 2.

The key question seems to be the true physical-chemical nature of the compact: is it straight distribution of the WSe_2 in the eutectic matrix (exothermic reactions indicate otherwise), or are intercalation compounds being formed in accordance with Professor Jamison's hypothesis?

The presence of any residual eutectic metal matrix may play an important role in the behavior of a high temperature solid lubricant. According to Shwin and his Russian coworkers (Reference 36), "one of the latest methods of obtaining heat-resistant materials is the development of composite materials with a metallic base reinforced with high-melting high-strength filaments or platelets. In contrast to precipitation-hardening and dispersion-hardening materials, composite materials reinforced with continuous or discrete filaments withstand substantial tensile stress due to the transfer of the force across the matrix-filament interface. Thus, heat resistance of the reinforcing filament is a requirement for the components of composite materials. The matrix of a composite material must be ductile at low temperatures and scale resistant.

Furthermore, no interaction should occur at the matrix — filament interface at high temperatures.

Eutectic alloys of the metal-interstitial phase system are natural composite materials with chemical compatibility, a strong bond between the filament and the matrix, and high structural stability at elevated temperatures (up to $0.9T_m$)."

This, of course, is the basic ideal of DuPont's (and now Stellite Division of Cabot Corporations') Tribaloy, a two-phase composition of hard, hexagonal Laves phase crystals tightly bonded by a softer matrix (Reference 37) and of other tribo-metallic compounds prepared by German researchers (Reference 38). MoS_2 has been known to be co-sputtered with a nickel-phosphorus eutectic providing improved wear resistance.

The above discussion is intended to show the variety of avenues open to the program, providing high temperature solid lubricants either in thin film form, or in solid compact for machining as a self-lubricating ball bearing retainer. The use of special tribo-metal alloys as bearing retainer stock is not discounted, with or without the added benefit of a low shear strength softcoat.

(c) Turbine Engine Bearing Retainer Preparation

The anticipated higher ball to ball pocket loads and higher speeds seen by the Type II bearings renders the layered, 2D graphite fiber reinforced, additiveless composite of the Type I bearing marginal, at best, for turbine bearing use.

In the present case, increased strength and reduced wear rate of the ball pocket contact areas and the race land contact zones are imperative. For this reason, 3D carbon/graphite weave reinforced, solid lubricant additive containing self-lubricating composite hoops will be designed to satisfy these requirements.

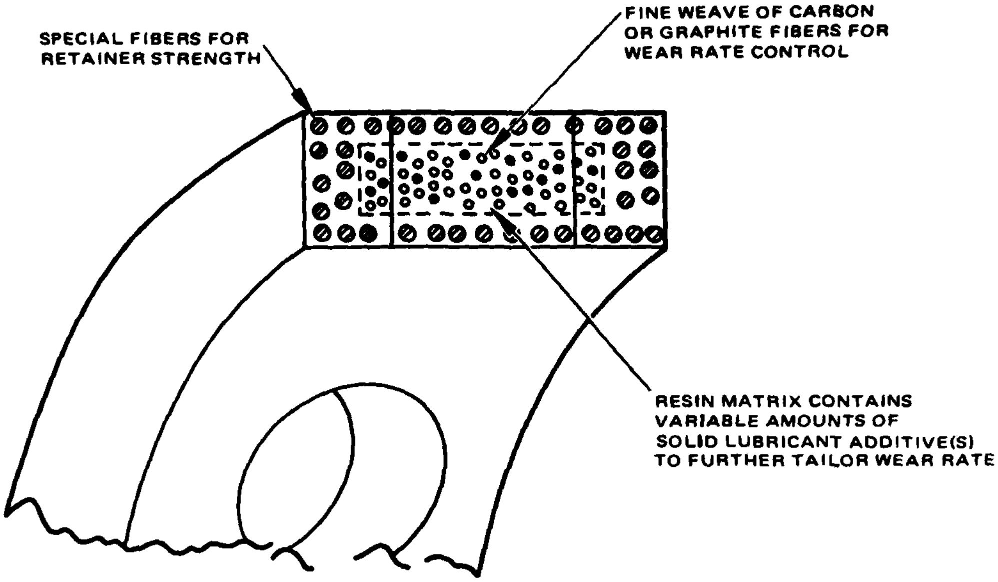
The portion of the work described here will deal with the preparation of 3D cylindrical preforms of carbon, graphite and carbon/graphite hybrid weaves by FMI to be impregnated with solid lubricant powder containing Thermid 600 polyimide by Hughes. The molded and cured composite tubes will be machined into ball bearing retainers of various strengths, stiffnesses, dynamic moduli, dimensional stabilities and wear rates by Hughes, mostly as a function of fiber type in the recommended preform design and the amount of lubricant additive in the resin.

Based on discussions conducted during the PDR, it was decided to abandon the "old" approach of preparing planar 3D weaves for preforms and wrapping these thick sheets around a mandrel to form a hoop. Instead, we switched to 3D cylindrical weaves, because weaving technology advanced to the point where the strength of the retainer, as well as its wear rate at the ball-to-ball pocket contact area, can be varied (and perhaps later, controlled) by appropriate weaving techniques. The basic idea is depicted in Figure 10, where the strength of the retainer, leading to high dimensional stability up to 600°F, is provided by (high modulus, high strength) fibers on the outside "skin" of the tube, while the wear rate in the ball pocket contact area is controlled by the type of carbon (G-10) or graphite (HMS) or carbon/graphite hybrid weaves placed in the middle of the tube. The relative fiber type and content of the 3D preform will first be predicted by structural analysis techniques at FMI.

Based on these predictions, we will attempt to vary the stiffness, modulus and other physical properties, as well as the wear rate and transfer film formation from the ball pocket by two methods: one supplied by FMI, the other by Hughes.

- Method 1 (FMI)

Weaving 3D cylinders which will contain carbon (G-10), graphite (HMS) or carbon/graphite hybrid weaves in the wear scar areas (see Figure 10) and reinforcing the retainer "skin" for extra stiffness



BEARING RETAINER DIMENSIONS:

RETAINER I.D. = 1.477 IN. / 1.467 IN

RETAINER O.D. = 1.626 IN / 1.620 IN

RETAINER WIDTH = 0.325 IN. / 0.320 IN.

BALL DIAMETER = 0.219 IN.

BALL POCKET DIA = 0.240 IN. / 0.234 IN.

NO. OF BALL (AND BALL POCKETS) = 17

Figure 10. 500°F turbine engine bearing retainer dimensions and basic idea of hybridized 3D cylindrical weaving of carbon/graphite fiber preforms for retainer rotational velocity of up to 30,000 rpm.

and dimensional stability with special (high strength, high modulus?) fibers. The actual fiber type should be predicted by FMI structural analysis and calculations and by the previous Hughes test results contained in those reports FMI already has. The preform cylinders must, of course, be slightly oversized, as compared to the retainer dimensions in Figure 10, to allow for molding and machining after molding.

- Method 2 (Hughes)

Impregnating these dense weaves with a Thermid 600 polyimide varnish, containing various percentages of our

advanced, high temperature solid lubricant additive (the powdered $Ga/In/WSe_2$ Westinghouse compact). The resin varnish should, but does not have to penetrate the anticipatedly dense weaves deeply, since the wear scar volume wear depth is small (see Appendix C).

"Dense" weaves and small tow sizes will be needed in order to provide good homogeneity of the ball pocket area's fiber content. In the case of hybrid weaves, the tow size must necessarily be small. In the case of a single fiber type in the wear scar area, the tow size may be somewhat larger. This argument is based on the data presented in Reference 39 and the calculations in Appendix C, done per the formulas given in Reference 40, and values guided by Reference 41.

The calculations involve estimations of the kind of ball/ball pocket contact areas and unit contact loads that may be anticipated, using known Hertzian stress formulas. For the sake of simplicity, only a one pound load is used in those calculations. Such loads, however, may be as high as five pounds (the exact values are not determined yet). Note that Professor Nypan at the California State University at Northridge has successfully used a special bearing retainer ball pocket load magnitude tester (see Reference 42) for fluid lubricated turbine engine bearings. Consultations with him revealed that this equipment could be modified to accept a smaller solid lubricated bearing of different configuration. Further analysis of the usability of this apparatus and proposed technique will be the deciding factor in Dr. Nypan's future involvement in our program.

One should be aware of the fact that there are practical problems to be faced in terms of weaving small tows, the availability of small tow sizes from the materials we need, etc. In the end, a compromise will have to be struck between what is needed and what can be done realistically. The test plan for the cooperative Hughes/FMI work is described in Appendix D.

Due to the specialized nature of the fairly thin walled hoop, the fabrication of separate friction and wear test specimens for intrinsic evaluation of tribological performance on simple, standard test machines is very difficult. Nevertheless, it will be attempted, especially for use in the ball bearing simulator (see forthcoming discussion).

(d) Turbine Engine Bearing Test Plan

The current state of the art of high load, high speed solid lubricated metal alloy bearings is a 1000°F, 20,000 rpm turbine engine bearing operating for ~50 hours (see Reference 43). These bearings operated with pressed-in, self-lubricating ball pocket inserts of the Ga/In/ WSe_2 Westinghouse compact. If one tries to surpass that speed, the separator begins to do vicious gymnastics that lead to its own (and the bearing's) destruction. (There is no guarantee that the substitution of e.g., Si_3N_4 for metal will cure the problem.) While Westinghouse-generated computer programs have significantly improved the wear life by extending the top limit to where it is today, new lubricant/cage materials development is needed before the next generation quantum jump can be made.

To achieve this (as mentioned before), the only reasonable approach at this time is to start with computer modeling to predict likely modes of instability for solid lubed versions of ball bearings, followed by the incorporation of advanced materials available at the start of the experimental work. The bottom line is to compare theory and practice under the same roof. The detailed portion of the MTI bearing test subtasks are described below.

- Task I - Test Rig Modifications: In this task an existing test rig will be modified to accept a rolling element bearing currently employed in a small high speed turbine engine.

An existing test rig presently installed in the MTI Seal and Bearing Test facility will be dedicated to the experimental

evaluation of a turbine engine, high load-high speed bearing. It is anticipated that one bearing size will be subjected to experiment so that only one design for the test head will be required. The selection of one bearing size will not limit the number of bearing designs investigated so long as the size envelope is not exceeded. The test facility will be modified according to the following tasks.

In the task a revised test head will be designed and constructed. The new test head will be installed in the test facility pictured in Figure 11. The general configuration for the revised head is shown in Figure 12.

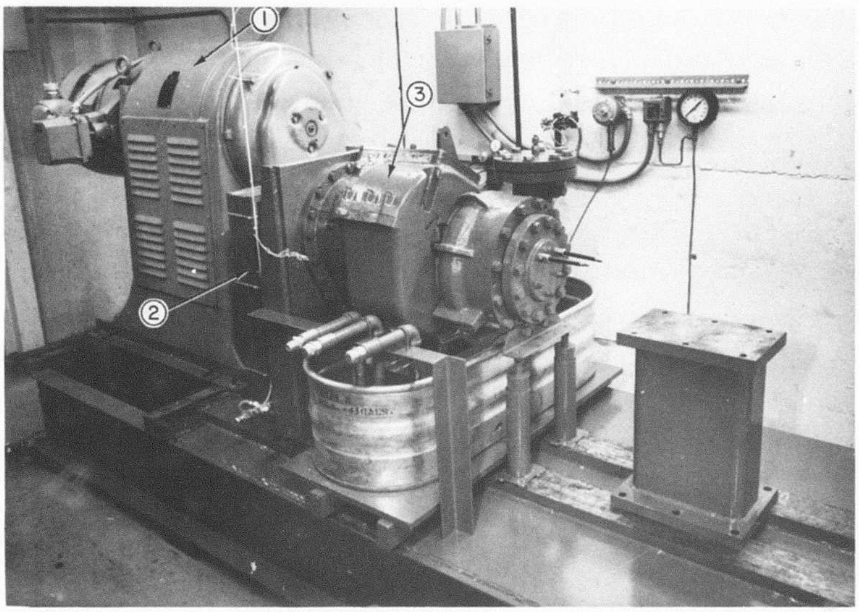


Figure 11. MTI turbine engine bearing test machine (for description of components, see text).

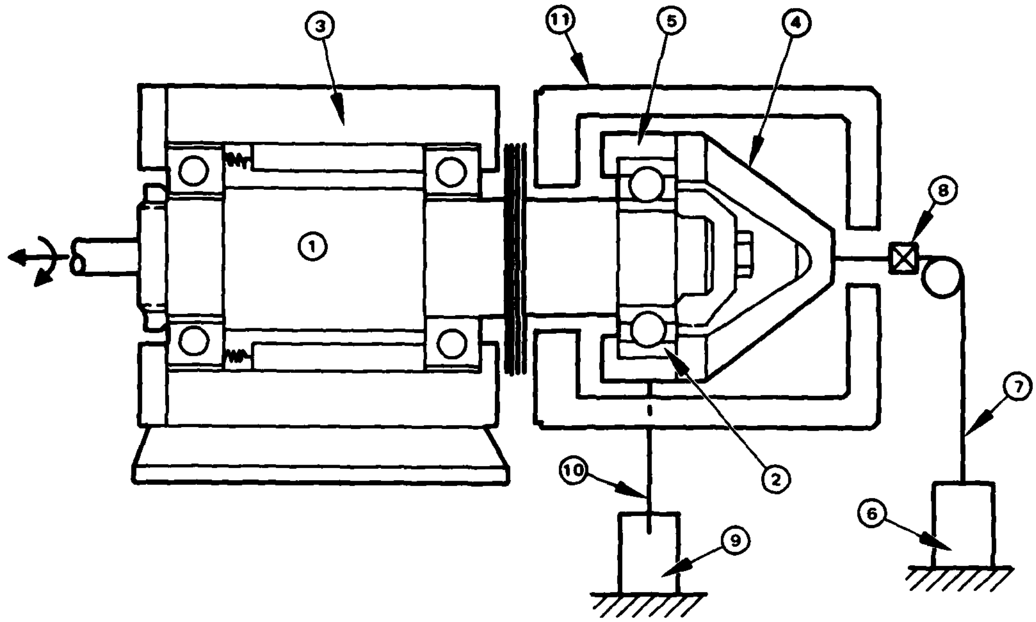


Figure 12. General schematic of the revised MTI turbine engine bearing test head for solid lubricated bearing tests (for description of components, see text).

The basic facility as illustrated in Figure 11 consists of a 75 HP variable speed drive (1) having a speed range of 450/1900 rpm. The prime mover is connected to a toothed belt transmission (2) which can provide gear ratios of 4/1, 1/1 or 1/4. The toothed belt transmission drives a step-up geared transmission (3) which has a ratio of 7.48. The output shaft of the transmission, therefore, will provide speeds over the range 840 rpm $\leq N \leq$ 56,000 rpm. The step-up transmission is a standard "Turbonetics"* gear box and is rated at 400 HP at 56,000 rpm. The new test head is pictured in Figure 12 and consists of the following components. At one end of a test shaft (1), a bearing seat is provided for the test bearing (2). A ball bearing spindle (3) provides the test shaft support. A tripod puller (4) applies the axial bearing load during a test sequence by pulling

*Division of MTI.

against the test bearing's outer seat ring (5). Actuation of the axial load is provided via a pneumatic cylinder (6) and a long, flexible cable (7). A guide bushing (8) is included in the axial load system to assist in its static support. Radial load is applied to the bearing via the pneumatic cylinder (9) through the load cable (10). An infrared heating oven (11) employing quartz electric heating elements will enclose the test bearing area to provide the necessary test temperatures.

It is assumed that only one bearing size will be tested, but that several designs fitting the same envelope may be tested. It is assumed that the test bearing envelope will not exceed three to four inches in diameter.

The performance envelope of the test rig will encompass the following:

• Rotating Frequency	To 56,000 rpm
• Axial Load	To 750 lb
• Radial Load	To 750 lb
• Test Temperature	To 1500°F

Actual test conditions will be established at the time of test with the concurrence of Hughes Aircraft Company.

A data acquisition system will be included in the test fixturing to provide data on the following:

- Ambient and outer race seat temperature
- Inner race rotational frequency
- Cage rotational frequency
- Cage axial motion

- Bearing running torque
- Duration of test

The acquisition of test bearings will be included in this task to assure appropriate delivery schedules commensurate with the program requirements. It is assumed that the test bearings to be acquired will include:

- Three (3) turbine engine bearings presently in use for modification with $Ga/In/WSe_2$ retainer inserts. The modification of the test bearings will be performed by MTI using detailed parts drawings in Reference 14.

- Thirty (30) test bearings of a new design to include sputtered Hughes composite and other retainers, i.e., with or without the use of an advanced high temperature compact, designed to replace the $Ga/In/WSe_2$, as a self-lubricating cage (insert) material. The complement of bearings will include:

  - M-50 test specimen components with two hardcoats.

  - M-50 test specimen components with two softcoats.

  - Si_3N_4 test specimen components obtained from AMFL free of charge, with or without special solid lubrication.

Perhaps improvements in the Si_3N_4 bearing surface after fabrication (going beyond the machining work of Professor Rabinowicz's, Reference 44, where the necessity of a final "integrity cut" was demonstrated) is the key. Surface integrity, even after employing the best machining techniques, may be improved by chemical vapor deposition (CVD) or sputtering with silicon nitride itself. Elimination of

surface microcracks by the above methods can then be followed by stabilization of the bearing surfaces against hot corrosion by other chemically resistant but compatible hardcoats and/or softcoats.

• Task II - Base Line Development: In this task test bearings containing retainers modified with Ga/In/WSe_2 ball pocket inserts will be tested. It is anticipated that two (2) bearings will be actually tested with a third bearing maintained as a backup in the event gross differences in test results are obtained. For these tests only one (1) level of speed, radial load, axial load and ambient temperature will be employed. Each bearing will be run to failure; the criterion of failure will be when the running torque increases by 30 percent from its initial value.

Each test bearing will be subjected to a pretest inspection which will include:

* Photographs of components (35 mm)

* Microscope photographs of selected locations on the inner races, outer races, cage pockets and balls. Two photographs of each race, and two pocket and two ball photographs will be taken.

Test data will be obtained from the following sensor installations:

* Ambient and outer race seat temperatures will be obtained from the output of three type K thermocouples imbedded in the outer race seat and one thermocouple suspended in the hot ambient space inside the heating oven.

* Inner race rotational frequency will be obtained by employing a magnetic pick-up which will serve six discontinuities machined onto the test shaft.

* Cage rotational frequency will be obtained from the output of an MTI high temperature capacitance probe which will view a surface discontinuity on the metallic bearing cage.

* Cage axial motion will be obtained from the same probe output as used for the cage rotational frequency. The measurement of cage radial motion will not be made.

* Bearing running torque will be measured from the strain of two diametrically opposed strain gage torque arms firmly attached to the test bearing outer race seat. The torque arms will penetrate the heater oven, permitting reduced temperature operation of the strain gages.

Data acquisition will be accomplished by multi-point chart recording of the test temperatures with one bearing outer race temperature recorded on a single pen strip chart. All other dynamic test measurements such as speed and torque will be recorded on magnetic tape.

A post test inspection will be performed on each failed bearing. This inspection will include sectioning, micrography and photography. It is expected that surface analysis of the failed bearing components will be performed at the Naval Research Laboratory.

- <u>Task III - Improved Bearing Design</u>: In this task the new design turbine bearings will be tested to failure using the same failure criterion as in Task II. The new design will be based on the results of the computer modelling study and all research dealing with the advanced bearing materials and their physical-chemical interaction with the candidate lubricants. Each bearing undergoing tests will be subjected to identical test conditions, consisting of one (1) radial load, one (1) axial load, one (1) rotational speed and one (1) ambient temperature to be specified at the time of test by MTI in accordance with DREB predictions, with concurrence of Hughes Aircraft Company.

Each data point will be run with two bearings resulting in a total of 16 tests. An additional data point will be taken using an indicated silicon nitride bearing, with or without special solid lubrication. The specimen will be supplied through an existing contract (see Reference 16).

Each test bearing will be subjected to a pre-test inspection which will include:

* Photographs of components (35 mm).

* Microscope photographs of selected locations on the inner races, outer races, cage pockets and balls. Two photographs of each race, and two pocket and two ball photographs will be taken.

* SEM survey of areas of which microscope photographs were taken.

A post test inspection will be performed on each failed bearing. This inspection will include: sectioning, micrography and photography. It is expected that surface analysis of the failed bearing components will be performed at the Naval Research Laboratory.

It is prudent to repeat here the manner of folding the results back into the DREB program:

- <u>Task IV - Evaluation of Experimental data:</u> At this point, the behavior predicted in the DREB tasks will be closely evaluated in the light of actual experimental data obtained within the scope of this project. This comparison may result in added modifications to DREB in order to make the analytical predictions more realistic. In such a case, the required update will be implemented in DREB and new runs will be made and compared with the experimental data. Such a close interaction between the

analytical predictions and the experimental observations will not only result in a realistic analytical design tool, but it will also help improve the general understanding of the dynamic behavior of solid lubricated ball bearings.

- Task V - Parametric Study: Once DREB is proven to be adequately sound in predicting the bearing behavior, a systematic parametric study over the entire operating range will have a significant design relevance. A number of computer runs will be made over the range of geometrical, materials and operating parameters and the general behavior of the bearings will be evaluated. In particular, bearing life as determined by cage instabilities, ball skidding, wear, fatigue, etc., will be investigated. As a result of this parametric evaluation, some design guides will be derived.

In addition to all MTI theoretical and practical tasks, technical assistance will be provided to the other participating companies with regard to the analysis of any experimental data. The wear data generated at AFML and Midwest Research Institute have already been identified as an example at the program PDR. These data require some statistical data analysis and possibly the development of a wear model. It is expected that as the project advances towards its goals, assistance in other similar areas will be mandatory in order to maintain close ties between the experimental work and the analytical modeling.

3. The Ball Bearing Simulator

Without the benefit of fully developed dynamic bearing computer programs (not yet available), the interaction between bearing parameters is so poorly understood that computer predictions alone would not be sufficient. For instance, on the one hand, a compliant hardcoat/softcoat layer can significantly reduce the contact stresses between the ball and the races in a ball (or roller) bearing. On the other hand, large localized build-up of film debris can seriously alter ball-race and ball-cage loading and will accelerate cage wear.

Unfortunately, bearing wear experiments with actual ball bearing specimens are useful for life-testing and qualitative specimen condition determinations only. Time-dependent composite wear and film formation phenomena cannot be observed because one cannot look inside of an assembled, solid lubricated ball bearing to measure lubricant film thickness and component wear. Periodic disassembly of "identical" bearings of a test set is not only expensive, but also unreliable because some of the wear-influencing parameters cannot be controlled accurately. For one, the individual ball/ball pocket loads are functions of the respective ball speed variations. Ball speed variation itself depends on component tolerances, bearing misalignment, and several other factors. There is always some variation in these factors from bearing to bearing, even if all of the test specimens were purposely selected to be "identical." The slightest of variations are cumulative and can induce wide scatter of ball pocket wear, and other relevant data.

In order to allow visual and instrumental observation of solid lubricant behavior under bearing-like conditions and in order to check the predictions of computer diagnostics and provide numerical values for dimensionless parameters, a special ball bearing simulator is needed to serve as an intermediate tool between the computer and the actual gyro and to some extent, the turbine engine bearing as well.

Simulator tests can check these predictions with respect to the ideal film thickness (as deposited), the upper tolerance limit for localized build-up and desirable wear rate retainer materials for a given load/speed/environment combination.

Battelle has undertaken the effort of developing an upgraded version of the design of Gardos and Preston (Reference 45), also guided by previous French work (Reference 46) and research performed by Battelle itself (Reference 47). The tester will be capable of testing (a) hardcoat/ sacrificial softcoat lubricated specimens, (b) transfer lubricated specimens and (c) (a)/(b) hybrid lubricated specimens.

The apparatus will contain the following features:

- Speeds. Race speeds up to approximately 5,000 rpm; oscillatory rate, 1800 cpm.

- Temperature. Up to approximately 250°F.

- Loads. Adjustable from zero to maximum Hertzian stress level of 350,000 psi.

Note that the oscillatory mode modification is incorporated, because we hope to conduct research with solid lubricated, oscillatory gimbal bearings (Type III) in the future. At this time, work on Type III bearings is beyond the scope of the program.

It is anticipated that the apparatus will consist of a ball loaded between two (20-30 mm diameter) bearing races. Ball size will be selected by the subcontractor and approved by Hughes. A segment of a transfer film separator will be incorporated into an adjustable holder, attached to an X, Y, Z transducer base capable of inducing controlled ball pocket loads and their measurements at the same time. This design allows free ball spin between the races, unlike any of the precursor apparatuses described in References 45 through 47.

Ball-to-race loads on the order of 200 pounds (900 N) will be required for the designed stress levels.

Battelie will supply and integrate into the test apparatus a special transducer and electronics for measuring the contact pressures between the ball and raceway specimen contacts. This technique involves the use of a vapor-deposited strip of the alloy manganin on top of a sputtered insulating-layer of alumina (Al_2O_3). The resistance of manganin is pressure sensitive. By measuring this resistance change as the manganin passes through contact between the ball and raceway in the bearing, a measure of

contact pressure is obtained. The life of this device will be as attainable with current technology (References 48 and 49).

Battelle will attempt to study the feasibility of and methods for the in situ measurement of the thickness of the solid film on the race specimen. Methods to be studied will include (but not be limited to):

(a) Optical methods

(b) Talysurf head in the fixture

(c) Capacitance detectors

(d) Thin film transducer pressure change.

Item (d) would of course involve empirical correlation of film thickness as measured with pressure transducer readings and some other standard.

The feasibility of and methods for implementation of ball separator pocket wear will also be examined.

Here, samples of the Hughes retainer components (Appendices B and D) and other candidates for retainer materials (lubricated by softcoats or self-lubricating compacts) can be screened with respect to wear behavior under bearing-like operation. Of course, due to initial design difficulties, a high temperature (i.e., above 250°F) version of the simulator will not be attempted until the present design will prove itself successful.

Examination of in-situ hardcoat/softcoat wear, self-lubricating retainer transfer film formation and film endurance, as well as visible race/ball wear resulting from solid lubricant failure will be conducted by a built-in, long focal length (100X magnification maximum) microscope and a Polaroid camera attachment.

The details of the pressure transducer development will depend on the bearing configuration(s) and operating conditions evolved in earlier tasks. The approach here will be to fabricate the transducers for selected bearings and to assist Hughes (on-site) in the operation of the transducer. Special provisions will be required in the bearing fixture to accommodate the transducer and the associated lead wire. Also, the operating conditions must be selected to optimize performance life of the transducer. The transducer electronics will be designed for readout on standard oscilloscope.

Since this fixture is based on a previously successful Hughes bearing simulator design, extensive Hughes participation in the present fixture design is affected by the involvement of Mr. Crawford Meeks of Hughes Aircraft Company, a recognized bearing expert and bearing test specialist in the field. Close liaison between Mr. Meeks and Battelle researchers will be maintained throughout all phases of this task.

Design layouts will be complete and of sufficient detail to be readily understood and to define shape of all parts and will specify all critical fits. Materials and processes that are important to proper functioning of the test apparatus will be specified on layouts.

Detail parts may be fabricated from sketches, multi-detail drawings or from design layout drawings. Sufficient engineering records will be maintained to permit duplication of any part in the assembly.

The target objective in the fabrication construction will be to achieve a self-contained portable unit. Ease and simplicity of operation will be stressed.

Battelle has recently completed a successful brass-board model of the simulator (see Figures 13 and 14). The preliminary behavior of the test specimens indicated the soundness of the basic design and resulted in the Hughes go-ahead for full design and construction.

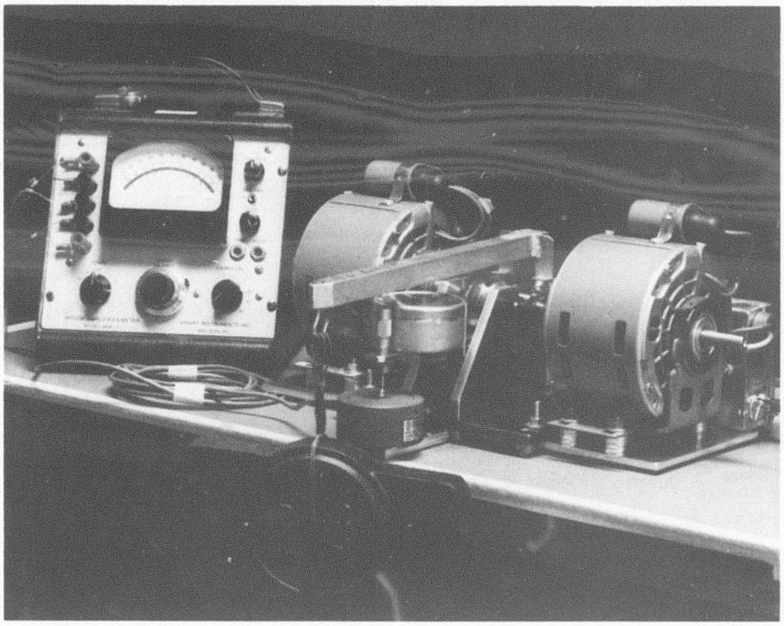


Figure 13. Overall view of Battelle brass-board model of ball bearing simulator.

At the completion of the final design the simulator will be delivered to Hughes Aircraft Company for shake-down and commencement of in-house testing of solid lubricant combinations.

As the next step, of course, the best materials and retainer concepts will be incorporated into the specific bearing types, leading to ideal bearing designs. Essentially, then, the DREB and other computer diagnostics, confirmed by the ball bearing simulator test results, will be translated into bearing/lubricant designs to produce useful hardware.

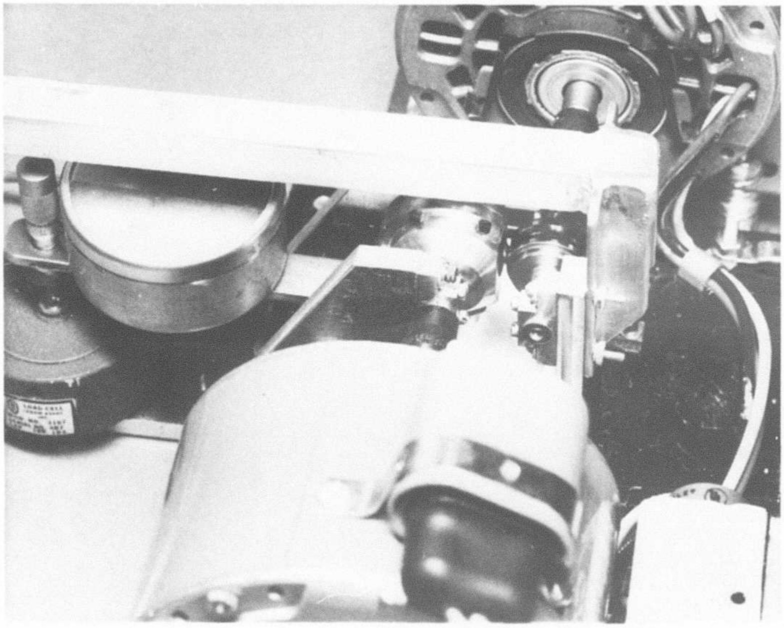


Figure 14. Test region of Battelle brass-board model of ball bearing simulator.

It should be noted here that during the second trip to Europe, we had the opportunity of meeting with Professor Horst Czichos and Dr. K.H. Habig of the Bundesanstalt fur Materialprufung (BAM), in West Berlin, Federal Republic of Germany. Professor Czichos and his team pioneered the systems analysis approach to tribological testing (References 50 through 52). Although all their previous work is confined to correlating meaningful laboratory testing and actual behavior of physical sliding tribo-systems, they have subscribed wholeheartedly to our approach to the ball bearing simulator. They felt that the closest possible approach to usable bench test data translatable to actual bearing operation is the incorporation of a simulator, like ours. This was also gratifying news to Mr. R.I. Christy of Hughes Aircraft Company, who is modifying his Faville No. 6 disc-on-disc

tester to incorporate the testing of ball bearing-like specimens. Mr. Christy will then attempt to correlate endurance tests of softcoats between the sliding disc-on-disc attachment and the rolling attachment.

IV. CONCLUSIONS AND FUTURE WORK

During this reporting period the DARPA/AFML/Hughes Solid Lubricated Rolling Element Bearing Program was put on a sound scientific basis. A program network was built, employing the experts in the field and allowing for flexibility because of the ability of the talent involved.

Forthcoming work will first include completion of all outstanding subcontracts, as shown in Table II. The timetable therein is considered reasonable if additional negotiations with DARPA are completed, as planned.

Meanwhile, Litton will continue evaluating variously machined bearing specimens and is preparing for gyro bearing baseline tests as soon as possible. MTI/SKF will accelerate the theoretical prediction work to arrive on level with the forthcoming Litton gyro bearing tests.

Parallel solid lubricants research will start, first with TMD, and in FY'80 with other research organizations (e.g., Westinghouse).

The immediately forthcoming Litton Phase I report (see Reference 2) will be a valuable supplement to this one, and will be distributed to the present recipients.

TABLE II. PRESENT SUBCONTRACT STATUS; TOTAL PROGRAM FINALIZATION ON CONCLUSION OF FURTHER NEGOTIATIONS WITH DARPA AND CERTAIN SUBCONTRACTORS.

U.S. FIRMS	UNDER CONTRACT	WORK OR CONTRACT REVISION NEEDED	TECHNICAL INFO FOR REVISION AVAILABLE	FINAL SOW* DUE DATE
BATTELLE	Yes	Yes	Yes	-
LITTON	Yes	Yes	Yes	-
MTI	Yes	Yes	Yes	-
BARDEN	Yes	Yes	Yes	-
FMI	No	Yes	Yes	15 May '79
SKF	No	Yes	Yes	28 Feb '79
MRI	No	No	Yes	28 Feb '79
TMD	No	No	Yes	15 May '79
TMI	No	Yes	No	15 May '79
WESTINGHOUSE	No	Yes	No	1 Feb '80

*Statement of Work

REFERENCES

1. Gardos, M.N., Hughes Aircraft Company Policy Letter to DARPA, AFML/MBT and the AFSC on planned DARPA Solid Lubricated Rolling Element Bearing Program, 22 September 1978.

2. Baginski, W., "The Present Tribo-Technical State-of-the-Art and the Development of a New Generation of Ball Bearings by Advanced Tribological Methods," Litton Guidance and Control Systems Report EWO 97704 to Hughes Aircraft Company, Contract No. 04-493431-FS5, Phase I, February 1979 (to be published).

3. Gass, H. et al, "Bearings for Extreme Environments," European Space and Technology Center (ESTEC) Report No. 95486, paper presented at the European Space Tribology Symposium, Frascati, Italy, 9-11 April 1975.

4. W. Ruppert, "Influence of Deposition Temperature on the Hardness and Dimensions of Steels with Hardcoatings Produced by Chemical Vapor Deposition," Thin Solid Films, 40, 27 (1977).

5. Mahoney, J.F., et al, "Development of a Table Top Model for the Generation of Amorphous/Microcrystalline Metal Powders", Report 78-1, Phrasor Technology, Duarte, California, Contract N00014-77-C-0373 (June 1978).

6. Personal communication with Professor W. Ruppert (also see Reference 2).

7. Gardos, M.N., "Quality Control of Sputtered MoS_2 Films," Lubrication Engineering, 32, 463 (1976).

8. Gupta, P.K., "Dynamics of Rolling Element Bearings, Part I: Cylindrical Roller Bearing Analysis", paper presented at the ASME-ASLE Joint Lubrication Conference, Minneapolis, Minnesota, October 24-26, 1978, ASME Preprint Paper No. 78-Lub-25.

9. Gupta, P.K., "Dynamics of Rolling Element Bearings, Part II: Cylindrical Roller Bearing Results", ibid., ASME Preprint Paper No. 78-Lub-26.

10. Gupta, P.K., "Dynamics of Rolling Element Bearings, Part III: Ball Bearing Analysis", ibid., ASME Preprint Paper No. 78-Lub-32.

11. Gupta, P.K., "Dynamics of Rolling Element Bearings, Part IV: Ball Bearing Results", ibid., ASME Preprint Paper No. 78-Lub-33.

12. Crecelius, W.J., and Pirvics, J., "Computer Program Operation Manual on *SHABERTH: A Computer Program* for the Analysis of the Steady State and Transient Thermal Performance of Shaft-Bearing Systems", SKF-AL76PO30, SKF Industries, Incorporated, 1976 (AFAPL-TR-76-90, AD-A042981).

13. Thompson, W.S., and Pirvics, J., "Methodology for Performance Prediction of a Helicopter Transmission under Oil Starvation", Proc. 3rd Biannual Aircraft Survivability Symp. sponsored by the Joint Technical Coordinating Group for Aircraft Survivability, Naval Post-Graduate School, Monterey, CA, 30 October - 2 November 1978.

14. Williams Research Corporation (WRC) Drawing Number D 29056, per WRC Specification F 7410; fabricated by The Barden Corporation, Code Identification Number 70854.

15. NASA Tech. Brief B75-10022, May 1975.

16. Scilingo, A.A., "Manufacturing Methods for Ceramic Bearing Components", SKF Industries, Inc., Report No. R-358-7-2, USAF Contract No. F33615-78-C-501, Interim Progress Report - 8/15/78 - 11/15/78, December 15, 1978, King of Prussia, Pennsylvania.

17. Anon., "Matériau Auto-lubrifiant à Haute Fiabilité", Journal du Frottement Industriel, 1978, No. 1, pp. 2 and 3, Centre Stephanois de Recherches Mécaniques, Hydromecanique et Frottement, St. Etienne, France.

18. Anon., "Frottement et Lubrification dans l'Industrie, IV - Traitments de Surface et Economies d' Energie", 69 Ecully, par les Chercheurs du Centre Stephanois de Recherches Mécaniques, Hydromecanique et Frottement, St. Etienne, France.

19. Magie, P.M., "A Review of the Properties and Potentials of the New Heavy Metal Derivative Solid Lubricants, "Lubrication Engineering, 22, 262 (1966).

20. Jamison, W.E., Ph.D. Thesis, U. of Cincinnati, 1969.

21. Jamison, W.E., and Cosgrove, S.L., "Friction Characteristics of Transition Metal Disulfides and Diselenides, "ASLE Trans., 14, 62 (1971).

22. Jamison, W.E., "Structure and Bonding Effects on the Lubricating Properties of Crystalline Solids," ASLE Trans., 15, (1972).

23. Jamison, W.E., "Electronic Effects on the Lubricating Properties of Molybdenum Disulfide and Related Materials, "ALSE Proceedings - 2nd International Conference on Solid Lubrication, 1978, pp. 1-8.

24. Kjekshus, A., and Pearson, W.B., "Phases with the Nickel Arsecide and Closely-Related Structures, Progress in Solid state Chemistry, 1, 83 (1964).

25. Jellinek, F., The Structures of the Chromium Sulphides, "Acta Cryst, 10, 620 (1957).

26. Wilson, J.A., and Yoffee, A.D., "Transition Metal Dichaleogenides - Discussion and Interpretation of Observed Optical and Structural Properties," Advances in Physics, 18, 193 (1969).

27. Jamison, W.E., Report to the Selenium-Tellurium Development Association, Greenwich, Conn., 1970.

28. Whittingham, M.S., and Gamble, Jr., F.R., "The Lithium Intercalates of the Transition Metal Dichalcogenides, "Mat. Res. Bull.," 10, 363 (1975).

29. Jamison, W.E., and Weber, H.S., "Sliding Load History Effects on the Friction of MoS_2 in Vacuum," ASLE Trans., 14, 156 (1971).

30. Kamarchik, P. Jr., and Margrave, J.L., "A Study of Thermal Decomposition of the Solid-Layered Flurocarbon, Poly (Carbon Monofluoride)." J. Therm. Anal. 11, 259 (1977).

31. Kamarchik, P. Jr., and Margrave, J.L., "Poly (Carbon Monofluoride): A Solid, Layered Fluorocarbon", Accounts of Chem. Res., 11, 296 (1978).

32. Gardos, M.N., and Castillo, A.A., "Self-lubricating Composite Bearing Materials", AFML-TR-77-69, Hughes Aircraft Company, May 1977.

33. Boes, D.J., "Unique Solid Lubricating Materials for High Temperature-Air Applications", ASLE Trans. 10, 19 (1967).

34. Boes, D. J. and Chamberlain, B., "Chemical Interactions Involved in the Formation of Oxidation-Resistant Solid Lubricant Composites", ASLE Trans. 11, 131 (1968).

35. Boes, D. J. et al, "The Solid Lubrication of Ball Bearings Under High Speed - High Load Conditions from -225 to +1000°F", Lub. Eng., 27, 150 (1971).

36. Shurin, A. K. et al, "Properties of Eutectic Alloys Containing Titanium Diboride at High Temperatures", Metallovedenie i Termicheskaia Obrabotka Metalloy (Metal Science and Heat Treatment), 19, 696 (1978), Kiev, Ukrainian SSR.

37. Cameron, C. B., et al, "Tribaloy Intermetallic Alloy Compositions - New Materials or Additives for Wear Resistant Applications," Proc. Nat. Powder Metallurgical Conf., Atlanta, Ga., May 13-14, 1975, Princeton, N.J., Metal Powder Industries Federation, 1975, pp. 41-51.

38. Rüdiger, O. and Exner, H. E., "Application of Basic Research to the Development of Hard Metals - Powder Metallurgical Review 9," Powder Metallurgy International, 8, 7 (1976).

39. Lancaster, J. K., "Geometrical Effects on the Wear of Polymers and Carbons", J. Lubr. Tech., Trans. ASME, 97, 187 (1975).

40. Carroll, J. G., "Contact Stresses in Lubricant Testers", Lubrication Engineering, 24, 359 (1968).

41. Goggin, P. R., "The Elastic Constants of Carbon-Fibre Composites", J. Mat. Sci., 8, 233 (1973).

42. Nypan, L. J., "Measurement of Separator Contact Forces in Ball Bearings Using a Derotation Prism", paper presented at the ASME - ASLE Joint Lubrication Conference, Minneapolis, Minnesota, October 24-26, 1978, ASME Preprint No. 78-Lub-35.

43. Personal communication with Mr. David Boes of Westinghouse R&D Center, Pittsburgh, Pennsylvania.

44. Rabinowicz, E., "Grinding Damage of Silicon Nitride Determined by Abrasive Wear Tests", Wear, 39, 101 (1976).

45. Gardos, M. N., and Preston, D. F., "A Ball Bearing Simulator - A Tool for Accelerated Testing of Self-lubricated Bearings", ASLE Trans., 20, 133 (1977).

46. Martin, C., et al, "Experimentation des Composites Auto lubrifiants pour Roulements Spatiaux," La Recherche Spatiale, 13, 20 (1974).

47. Snediker, D. K., "An Apparatus for the investigation of Lubricant Transfer Mechanisms in Low-Speed Oscillatory Bearings," Wear, 30, 135 (1974).

48. Kannel, J. W., et al, "A Method for Measuring Surface Temperature between Rolling/Sliding Steel Cylinders", paper presented at the ASLE-ASME Joint Lubrication Conference, Kansas City, Missouri, October 3 - 5, 1977, ASME Preprint No. 77-Lub-1.

49. Fujioka, N., et al, "Electrical Resistance of Manganin Under High Static Pressures", J. Appl. Phys., 49, 4830 (1978)

50. Czichos, H. and Mølgaard, J., "Towards a General Theory of Tribological Systems", Wear, 44, 247 (1977).

51. Habig, K.H., "Tribological Tests on Surface Layers Formed by the Diffusion of Nitrogen, Sulfur, Tin, Boron or Vanadium into Steel," Z. Werkstofftech., 8, 267 (1977).

52. Czichos, H., "Tribology and the Reliability of Technical Systems", Materialprüf., 20, 33 (1978).

APPENDIX A

PRELIMINARY SPECIFICATION FOR R-3 TYPE, 440C, CrC-TiC COATED BEARING

APPENDIX A

PRELIMINARY SPECIFICATION FOR R-3 TYPE, 440C, CrC-TiC COATED BEARING

1. Dimensions - Tolerances - Finishes for Races and Balls

Dimensions shall be as specified in the following:

(In case tolerances are not specified, tolerances for A ABEC 7P shall apply).

Type:	Angular Contract, B type (inner race separable).
Size:	R-3 type, inch series, 8 balls
Class:	ABEC 7P or better (AFBMA standards)
Ball Diameter:	Specified elsewhere (nominal 2.381 ±0.013 mm). Balls assembled into one bearing shall have the same diameter within ±0.1 μm tolerance.
Outside Diameter:	0.5000 ±0.0004 in (12.7 ±0.01 mm)
Bore Diameter:	0.1875 ±0.0002 in (4.763 ±0.005 mm)
Width (Inner and Outer Race):	$0.1562 \pm^{0.000}_{0.001}$ in ($3.967 \pm^{0.000}_{0.025}$ mm)
Land Diameter:	
Outer:	0.4120 ±0.0005 in (10.465 ±0.013 mm)
Inner:	0.271 ±0.0004 in ($6.880 \pm^{0.0}_{0.01}$ mm)
Pitch Diameter:	0.3398 Ref. (8.63 mm)
Inner-Race Stick Out at 2.0 lbs:	-0.0033 to +0.0007 in (-0.160 to +0.18 mm)
Cross race radius of curvature:	57 percent for both, radius tolerance of above in 18° to 28° contact zone ≤ 0.000016 in (0.4 μm) Goal; 0.000024 in (0.6 μm) Max

Contact angle from radial plane at 2 lbs axial load:	23 ±2 degrees
Roundness of race tracks	
Inner:	0.000020 in (0.5 μm)
Outer:	0.000040 in (1.0 μm)
Race track surface finish:	0.000001 in RMS (0.025 μm RA)
Outer race land area finish:	0.00001 in RMS (0.25 μm RA)
Ball Classification (AFBMA standard):	Class 3

2. Retainers - supplied by Hughes/Barden/Litton team.

3. Materials - Coatings

3.1 Base material for races and balls:
AISI 440C, double or single vacuum melted, if former not available

3.2 Coating on inner and outer races:
CVD-titanium carbide/chromium carbide solution treated diffusion coating

Recommended total coating thickness: 0.0002 ±0.00008 in (6 ±2 μm)

Minimum finished coating thickness: 0.00012 in (3 μm)

3.3 Lubricant: SRG 160, to be supplied by Litton; for use in quality control tests.

4. Processes

4.1 Cleaning: per GA 300 (Litton) or a documented equivalent RMB procedure.

4.2 Races with optimum geometric conditions should be matched for optimum quality.

5. Quality Provisions

5.1 Barstock to be selected will be checked for hardness and should be R_c62 or better. Approximately 3 percent of the samples rejected for geometry defects shall be taken and sent to Litton for analysis.

5.2 Race track surfaces to be finished to optimum surface integrity, exhibiting the minimum residual stress condition. Samples provided to Litton as per 5.1 will be examined for surface conditions.

5.3 Wettability checks after final grinding shall be performed according to 5.3.1 to 5.3.3 on 3 percent of deliverable bearing races.

5.3.1 Wettability of metal parts. The entire surface of the race grooves and balls shall be finished, cleaned and packaged so as to provide and preserve the ability of the lubricant to spread on these surfaces.

5.3.2 Acceptable race wettability is defined as coverage of at least 8 percent of a race-groove surface in less than 17 hours after application of an 80 ±10 microgram drop of SRG-160 lubricating oil to the dry race. The drop may be applied to any part of the race. The race shall be oriented to locate the drop in the uppermost position with the race axis horizontal. The test shall be performed in an environment conforming to Fed-Std-209 Class 10,000 or better.

5.3.3 Ball surfaces should be uniformly wetted by the SRG 160 oil. Surface contamination, causing retraction or beading of oil film is unacceptable.

6. <u>Quality Assurances</u>

6.1 100 percent inspection and documentation per RMB data sheets for * required for each delivered bearing.

6.2 Original traces of 100 percent inspected - out-of-roundness - cross race curvature - surface finishes on race tracks are required for each delivered bearing.

6.3 10 percent of the bearings to be delivered shall be low speed torque tested (CW, CCW) per established and documented RBM procedure.

6.4 One percent sampling for outer race land surface roughness measurement is required.

6.5 Each delivered bearing shall have the actual ball size recorded. Two spare balls of the same size shall be provided.

7. Optional tests: established and documented axial compliance checks per RMB method No. M233 shall be performed on 10 percent of the delivered bearings.

8. General Considerations

8.1 This bearing will be used in an instrument which demands a low vibration, low torque and variations of the above.

8.2 These bearings are designed as test specimens to demonstrate superior wear characteristics using advanced lubricant systems.

*Outer rings, inner rings and bearing.

APPENDIX B

GYRO BEARING RETAINER PREPARATION PLAN

APPENDIX B

GYRO BEARING RETAINER PREPARATION PLAN
(DARPA/AFML Hughes Solid Lubricated Rolling Element Bearing Program)

I. SCOPE

Design tool(s) and develop fabrication method(s) for rosette-overlap tubes of layered carbon/graphite and Kevlar fabrics, impregnated with Thermid 600 polyimide resin. Since the initial goal is to develop high modulus, high hoop strength, dimensionally stable retainers that need not provide transfer lubrication, the tube materials will not contain any lubricant additives (e.g., MoS_2/Sb_2O_3 or the powdered Westinghouse composite).

II. MATERIAL

Carbon, graphite and Kevlar 2D weave reinforced Thermid 600 tubes.

III. REINFORCEMENTS

A. Thornel 300 fabric (4.5 mils thick) by Fiberite, Inc.

B. HMS fabric (8.0 mils thick) by Textile Products

C. G-10 or CCA3 fabric (7.0 mils thick) by HITCO

D. Kevlar fabric (3.0 mils thick) by Clark-Schwebel Fiberglass Corp.

IV. TUBE DIMENSIONS

OD	0.500 inch
ID	0.200 inch
length	6.0 inches (nominal)

V. APPLICABLE R-3 RETAINER DIMENSIONS

OD	0.475 inch
ID	0.310 inch

VI. GENERAL WORK PLAN

Prepare a minimum of three good tubes per reinforcement type by preparing a "test-tube" and cross-sectioning it at several places along its length to determine porosity and dimensional homogeneity. If the first tube is OK, prepare three others for machining by Barden. When all tubes have been fabricated and machined into retainers, do hoop strength and compression tests at Hughes. Pick two tube types: the best and the worst. Prepare test flats from layered flats equivalent of the two tube types. Machine a gamut of physical property test specimens from both flats and test (i.e., microtensile, microcompression, MRI rubshoe and tentatively, Rheovibron specimens), as described below.

VII. TUBE/FLAT MATERIALS PREPARATION AND TESTING

It is our intention to prepare both tubes and some flats of the layered materials and fabricate specimens from both configurations.

Physical Testing Specimen Types

1) From Flats - Fabricate from each designated material (as described above) two each microtensile (1/8 inch test width) (ASTM D638); two each microcompression specimens (ASTM D695-69)(0.3 x 0.3 x 0.6 inch); eight each MRI rubshoes (four bare, four MoS_2 sputtered); and one each Rheovibron specimen (tentative). Machining directions with respect to direction of molding and the consequent directions of test force applications will approximate directions and real forces in actual retainers.

2) From Tubes - Use Barden fabricated R-3 gyro retainers for test specimens, as needed.

Friction and wear test specimens of the layered flat stock shall be forwarded to MRI, Kansas City, Missouri for comparison with work previously performed (see Reference 32) and to AFML/MRI, Wright-Patterson AFB, Ohio for wear equation work. Special versions of the flat stocks will contain various amounts of solid lubricant additives, even though the Type I bearing retainer stock will not contain any. The reason is explained below.

All the above tribological testing work shall be done at both room temperature and at elevated temperatures (500° and 600°F). Since the turbine engine bearing hoops do not lend themselves to such specimen preparation (also see Appendix D), the 2D reinforced versions, prepared with the same fibers the 3D weave preforms will consist of will provide some guidance with respect to correlated tribological performance.

VIII. TIME FRAME OF WORK PLAN

January, February and March 1979: prepare four different types of tubes, machine retainers out of them at Barden and test retainer samples for hoop strength and compression set (dimensional stability). In April, May and June 1979: make additional tubes and equivalent flats of the best

and the worst materials, machine retainers out of tubes or save tubes for machining at some future date, and fabricate test specimens from flats and test the flat specimens. Deliver "best" and "worst" special retainers to Litton as soon as samples are available. These samples, along with the baseline cotton or paper reinforced phenolic retainers shall be sputtered with MoS_2 or other solid lubricant prior to gyro bearing assembly and test.

Deliver bare and sputtered flat stock friction and wear specimens to MRI and AFML/MRI for further testing and measurements.

APPENDIX C

GARDOS, M.N., BALL TO COMPOSITE BALL POCKET CONTACT AREA AND LOAD ESTIMATES

(Private research notes, 20 December 1978, Hughes Aircraft Company, Culver City, CA.)

APPENDIX C

BALL TO COMPOSITE BALL POCKET CONTACT AREA AND LOAD ESTIMATES

(Private research notes; M.N. Gardos; Hughes Aircraft Co.)
20 December 1978

1. Sphere-on-Spherical Seat (see Attachment 5)

$$a' = \left[\frac{(0.75)(\pi)(P)(k_1 + k_2)(R_1)(R_2)}{(R_1 + R_2)} \right]^{1/3}$$

where

a' = Hertzian half-width, i.e., the radius of the contact circle

π = 3.1415

P = ball-to-spherical seat load

k_1, k_2 = constants based on elastic modulus and Poisson's ratio of the respective contacting materials

$$\left[k = \frac{1 - \upsilon^2}{\pi E} \right]$$

R_1 = radius of ball (positive value)

R_2 = radius of seat (negative value); note that $R_2 > R_1$ always, so $(R_1 + R_2)$ is always negative, so therefore the ratio a' is always positive.

also

$$k_1 \text{ (composite seat)} = \frac{1 - (0.40)^2}{(\pi)(4 \times 10^5)^*} \cong 6.7 \times 10^{-7}$$

$$k_2 \text{ (steel ball)} = \frac{1 - (0.30)^2}{(\pi)(30 \times 10^6)} \cong 0.7 \times 10^{-9}$$

*conservatively low value

and

$$q'_o = \frac{3P}{2\pi a'}$$

where q'_o = Hertzian stress, i.e., the unit load; the other parameters as before.

If $P \cong 1$ pound, $a' \cong 1.30 \times 10^{-2}$ in.; $q'_o \cong 2800$ psi.

2. Sphere-on-Plane ($R_2 = \infty$)

$$a'' = \left[(0.75)(\pi)(P)(k_1 + k_2)\, R_1\right]^{1/3}$$

$$q'' = \frac{3P}{2\pi a''}$$

If $P = 1$ pound, $a'' \cong 5.59 \times 10^{-3}$ in.; $q''_o \cong 15{,}000$ psi.

3. Sphere-on-Cylindrical Seat

$$a \cong \frac{a' + a''}{2} = \frac{(1.30 \times 10^{-2}) + (5.59 \times 10^{-3})}{2} = 9.3 \times 10^{-3} \text{ in}$$

$$q_o = \frac{q'_o + q''_o}{2} = \frac{2{,}800 + 15{,}000}{2} = 8{,}900 \text{ psi}$$

A Hughes computer program predicted that, in this case, the contact (projected) area is elliptical, very close to a circle, where

$$\frac{a}{b} = 1.229,$$

so the contact area (projected) is $ab\pi = 1.229\, b^2\pi$.

Per Hughes' past experience with Teflon based composite retainers, b may be as large as one-half of the wear scar width $\cong$ 1 mm $\cong$ 3.937×10^{-2} in. So the projected area is $1.229\ b^2\pi = 5.98 \times 10^{-3}$ in^2, leading to a 167 psi load (P = 1 pound).

On the onset of ball to ball pocket sliding, the wear scar width rapidly grows from about 1×10^{-2} to 4×10^{-2} inches, with a corresponding quick drop in ball to ball pocket loads from about 9000 psi to less than 200 psi.

This first order analysis should help in visualizing the magnitude of these forces and the small wear scar areas that need to be "homogenized" by dense weaving.

APPENDIX D

TURBINE BEARING RETAINER PREPARATION PLAN

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TURBINE BEARING RETAINER PREPARATION PLAN

(DARPA/AFML/Hughes Solid Lubricated Rolling Element Bearing Program)

Step 1. FMI performs the preliminary structural analysis based on a previous FMI/Hughes meeting, previous Hughes research reports, and other information provided by the Hughes Program Manager. FMI recommends 3D cylindrical weave forms, tow sizes and fiber types for the reinforcing "ribs" or "skin" of the tubes and prepares another technical discussion and statement of work dealing with the upgraded preforms.

Step 2. The Hughes PO will be issued. FMI works out the practical details of preparing a foot long (or longer) tubular preform, starting with the ideal "rib-skin" reinforcement fiber and with G-10 in the ball pocket wear scar area. FMI delivers one tube.

Step 3. Hughes impregnates the preform with the solid lube containing resin varnish, dries the preform on a mandrel and rough-cuts rings of the prepreg. The rings are cured either in an autoclave while placed on a mandrel or in a special hot mold designed for low radial pressure on the composite hoop. This mold is needed more to keep the hoops in-round during the curing process than to press the thin wall of the prepreg for providing a "pore free" cured hoop wall structure. After cure, the hoops are final machined into actual retainers and other test specimens, and test results are obtained.

Step 4. During the Hughes effort described in Step 3, FMI prepares a reinforced "rib-skin" tube with HMS fibers in the ball pocket contact area and delivers one tube to Hughes for preparation and testing, as in Step 3.

Step 5. Hughes obtains test results on both tubes. Hughes and FMI jointly evaluate the results and make decision on the preparation of a reinforced "rib-skin" tube with a hybrid G-10/HMS weave at the ball pocket contact area. The decision will probably be on the relative surface areas of the respective exposed fibers in the wear scar area and not whether we will or we won't prepare the hybrid weave.

Step 6. FMI prepares several tubes of the best type(s).

It has been jointly agreed that stepwise preparation of the preforms is best, where we learn as we go. In addition to the information given in Figure 10, note that the density of the advanced Hughes 3D VYB 70 1/2-reinforced Thermid 600 polyimide composite (see Reference 32) is 1.52 g/cm^3 (resin-additive content ~36 w/o); the density of our 3D Thornel 50-reinforced version is 1.38 g/cm^3 (resin-additive content of ~ 34 w/o due to the tighter weave). The density of the pure Thermid 600 resin is 1.36 g/cm^3. Hughes generated no tensile strength values on these composites, but the tensile strength of MoS_2/Sb_2O_3-filled (70 w/o) Thermid 600 is ~ 4000 psi at room temperature. FMI will use these values to estimate the stresses within a spinning composite hoop with holes drilled into it. Superimposed to those stresses are the ball speed variation caused changes in ball-to-ball pocket loads: during bearing operation, certain balls can slightly advance and "push" against the ball pocket and others lag behind, being dragged along by the ball pocket of the retainer.